TP 10049E

STATE OF THE ART REVIEW OF PRESSURE RELIEF DEVICE DESIGN, TESTING, AND MODELING

Prepared for
Transportation Development Centre
Policy and Coordination Group
Transport Canada
August 1989

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Department of Mechanical Engineering

Queen's University

Kingston, Ontario, Canada

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92-00970/

The contents of this report reflect the views of the authors and not necessarily the official view or opinions of the Transportation Development Centre of Transport Canada.

Un sommaire en français de ce rapport est inclus avant la Table des matières.

Transport Canada / Transports Canada

PUBLICATION DATA FORM

1. Transport Canada Publication No.	2. Project No.	3. Recipient's Catalogue No.
TP 10049E	7215	

4. Title and Subtitle	5. Publication Date
STATE-OF-THE-ART REVIEW OF PRESSURE RELIEF DEVICE DESIGN, TESTING, AND MODELING	August, 1989
	6. Performing Organization Document No.
7. Author(s)	8. Transport Canada File No.
P.M. Petherick and A.M. Birk	D1465-302-6-7
9. Performing Organization Name and Address	10. DSS File No.
Department of Mechanical Engineering McLaughlin Hall, Queen's University Kingston, Ontario K7L 3N6	14SD.T8200-8-8555/A
	11. DSS or Transport Canada Contract No.
	T8200-8-8555/01-SD
12. Sponsoring Agency Name and Address	13. Type of Publication and Period Covered
Transportation Development Centre (TDC) 200 René-Lévesque Blvd. West West Tower, Suite 601 Montreal, Quebec H2Z 1X4	Tech.:Feb.-Sept.1989
	14. Project Officer
	D.W. Dibble

15. Supplementary Notes (Funding programs, titles of related publications, etc.)

16. Abstract

It is well known that the response of a rail tank car to exterior heating (eg. fire engulfment) is significantly affected by the operating characteristics of the pressure relief valve (PRV). If the valve jams or fails in some way, it can lead to a violent vessel rupture; therefore, relief valve failure modes and mechanisms must be understood.

This report results from an in-depth literature search to find data and methods for the analysis of pressure relief valves such as those used on rail or highway tank cars. Recent developments in relief valve design are identified. Data and models are presented and research needs are identified.

Specific areas which are considered in this report include: 1) sticking or jamming of PRVs; 2) ambient or fire effects on PRV performance; and 3) effects of age on PRV performance.

17. Key Words	18. Distribution Statement
Rail Tank Car, Explosion, Rupture, Pressure Relief Valve	Limited number of copies available from the Transportation Development Centre

19. Security Classification (of this publication)	20. Security Classification (of this page)	21. Declassification (date)	22. No. of Pages	23. Price
Unclassified	Unclassified	-	xvi, 82	-

TDC/CDT 79-005
Rev 88

Canada

Transports Canada Transport Canada

FORMULE DE DONNÉES POUR PUBLICATION

1. N° de la publication de Transports Canada	2. N° de l'étude	3. N° de catalogue du destinataire
TP 10049E	7215	

4. Titre et sous-titre	5. Date de la publication
STATE-OF-THE-ART REVIEW OF PRESSURE RELIEF DEVICE DESIGN, TESTING, AND MODELING	Août, 1989
	6. N° du document de l'organisme exécutant

7. Auteur(s)	8. N° de dossier — Transports Canada
P.M. Petherick et A.M. Birk	D1465-302-6-7

9. Nom et adresse de l'organisme exécutant	10. N° de dossier — ASC
Department of Mechanical Engineering McLaughlin Hall, Queen's University Kingston, (Ontario) K7L 3N6	14SD.T8200-8-8555/A
	11. N° de contrat — ASC ou Transports Canada
	T8200-8-8555/01-SD

12. Nom et adresse de l'organisme parrain	13. Genre de publication et période visée
Centre de développement des transports (CDT) 200 ouest, boul. René-Lévesque Tour ouest, suite 601 Montréal, (Québec) H2Z 1X4	Tech.:fév.-sep.1989
	14. Agent de projet
	D.W. Dibble

15. Remarques additionnelles (Programmes de financement, titres de publications connexes, etc.)

16. Résumé

Il est bien connu que le compartement d'un wagon-citerne soumis à une source de chaleur extérieure (par exemple un incendie enveloppant) est grandement conditionné par les caractéristiques de fonctionnement de la soupape de sûreté. Le blocage ou une défaillance quelconque de cette soupape peut provoquer la rupture violente du réservoir; par conséquent, il importe de bien comprendre les mécanismes et modes de défaillance des soupapes de sûreté.

Ce rapport est le résultat d'une recherche documentaire en profondeur visant à trouver des données et techniques d'analyse du comportement des soupapes de sûreté installées sur les camions ou les wagons-citernes. Les plus récentes améliorations apportées aux soupapes de sûreté y sont indiquées. Différents modèles sont présentés, fiche technique à l'appui, et les besoins de recherche y sont identifiés.

Les points spécifiques étudiés dans ce rapport incluent: 1) le blocage ou le grippage des soupapes de sûreté; 2) les effets des conditions ambiantes ou du feu sur le fonctionnement des soupapes de sûreté; et 3) les effets du vieillissement sur le comportement des soupapes de sûreté.

17. Mots clés	18. Diffusion
Rail, wagon-citerne, explosion, rupture, soupape de sûreté.	Le Centre de développement des transports dispose d'un nombre limité d'exemplaires.

19. Classification de sécurité (de cette publication)	20. Classification de sécurité (de cette page)	21. Déclassification (date)	22. Nombre de pages	23. Prix
Non classifiée	Non classifiée	-	xvi,82	-

CDT/TDC 79-005
Rev 88

Canada

SUMMARY

Pressure vessels such as rail and highway tank cars carrying pressure liquefied gases (PLG's), such as propane, are usually equipped with a pressure relief valve (PRV). These devices are designed to release the contents of the tank in the event that an uncontrolled pressure increase occurs (eg. if the tank is accidentally exposed to fire).

On occasion, these PRVs have been observed to malfunction, resulting in their slow response, partial opening, failure to reclose, or, in some rare cases, failure to open at all. In the most serious cases, the malfunctions can lead to the uncontrollable rise of the tank's internal pressure. Under controlled fire tests, these increased pressures have been shown to cause the catastrophic failure of tanks. Because of the serious consequences of a PRV malfunction, it is important that PRV operating characteristics be understood.

The objective of the present study was to determine the current state of knowledge of PRV technologies. The study involved a literature search and review in the area of PRV technologies. The original focus of the study was to conduct a literature search to find the state-of-the-art for PRVs which are currently being installed on railway tank cars, highway tankers, and stationary storage vessels. When few papers were found that had concentrated on the topic, the authors continued the search by considering both the nuclear power and chemical processing industries, where similar technologies were found.

The information was obtained by conducting a manual search of public domain literature. After screening, papers deemed particularly relevant were placed into one of three categories: 1) experimental studies; 2) modeling studies; or 3) case studies.

As near as could be determined, the design of PRVs currently installed on tank cars and highway tankers dates back more than 30 years. Attempts to discover any current variations were unsuccessful because of the proprietary nature of the information. Generally it is difficult to obtain detailed data on PRVs from manufacturers (beyond what is published in the PRV catalogues).

The majority of the experimental research on PRVs installed on rail tank cars has been conducted at the University of Maryland under the direction of D.W. Sallet.

Such topics as flow patterns within the PRVs under both single and two-phase flow conditions and tank depressurization phenomena have been addressed. None of Sallet's tests, however, have considered a PRV that was free to move. In each case, the seat was set at a pre-established height. This limitation has prevented the obtaining of valve dynamic characteristics. These are important bacause valve dynamics can play a key role in the life of a PRV.

A number of computer models of PRVs were located. However, few modeling studies of PRVs have compared the analytical results obtained with actual test data. Those that have, continually observed a poor correlation between theoretical and experimental data. Another problem with the valve dynamic models was that they were generally mated with simplistic source models (ie. the source is the system that the PRV is designed to protect, eg. tank-car in a fire).

The case study literature located was primarily associated with maintenance programs. No accident data was found that provided detailed information on PRVs. Investigations have shown that regular maintenance testing programs are seldom followed. Where limited testing is conducted, in both the nuclear power and chemical processing industries, results indicate that either many minor improvements in design, quality control, and application are needed, or else that fundamental rethinking is called for in PRV technologies. Field experiences show that PRVs often fail to meet specifications. Frequently, flow conditions within the valve are observed to result in chatter. This action can damage the valve, resulting in such effects as jamming, leakage, and, in some cases, the failure to open at set pressures. In some cases, even brief operation at the set point has been observed to cause damage to the PRV.

Very little information was cited which related directly to accidents caused by PRV failures. In the majority of cases, the incident which originally led to the rise in pressure was listed as the cause of the accident, and no mention was made of the PRV operation. The data which does exist on PRV performance during releases, obtained during controlled fire tests of tank cars, suggest that they often function poorly. In the most serious of cases it has been observed that their malfunction has caused the violent rupture of the tank.

During the course of the literature search, several research needs were identified. A summary of these is presented below:

1) Experimental research of valve operating characteristics are required. This is needed to better understand the dynamics of PRV operation.

2) Maintenance and inspection programs in the nuclear and chemical industries are insufficient. The usual procedures employed in the transportation of dangerous goods should also be examined.

3) Validation of existing dynamic computer models for PRVs is required. These validated codes should be mated with appropriate source models (such as thermal models of tank-cars) and used in future design improvement studies.

4) When an accident involving a pressure vessel occurs, the PRV should be removed and tested extensively to determine whether or not it performed correctly. This data should form a part of any accident report.

5) A data base should be established for accidents involving PRVs.

SOMMAIRE

Toutes les enceintes - wagons ou camions-citernes - servant au transport des gaz liquéfiés sous pression, comme le propane, sont protégées par des soupapes de sûreté. Celles-ci sont conçues pour laisser s'écouler la cargaison dès que se produit une surpression imprévue due, par exemple, à un feu extérieur.

Il arrive parfois que ces soupapes fonctionnent mal: lenteur d'action, ouverture partielle, incapacité de se refermer. À l'occasion, elles ne fonctionnent pas du tout. Ces défauts de fonctionnement peuvent, dans les cas les plus graves, entraîner des surpressions incontrôlables à l'intérieur de l'enceinte qui, comme l'ont montré les essais dans des conditions contrôlées, entraînent à leur tour la rupture brutale de l'enceinte avec des conséquences catastrophiques. Cela étant, il est impératif que les caractéristiques de fonctionnement de ces soupapes soient approfondies.

La présente recherche a eu pour but de faire un tour d'horizon des techniques connues concernant les soupapes de sûreté: recherche documentaire et analyse des technologies existantes, notamment en ce qui concerne celles destinées à protéger les wagons et camions-citernes et les réservoirs fixes. Les auteurs, ayant constaté le peu de communications traitant de ce sujet en particulier, ont décidé de se tourner vers les secteurs du nucléaire et des procédés chimiques où des techniques analogues sont mises en oeuvre.

L'information voulue a pu être obtenue grâce à une recherche par des moyens non automatiques des communications tombées dans le domaine public. Après filtrage, les communications jugées particulièrement intéressantes ont été classées dans l'une des trois catégories suivantes: 1) études expérimentales, 2) modélisation ou 3) cas d'espèce.

Le modèle des soupapes couramment utilisées pour la protection des wagons et des camions-citernes remonte, autant qu'il a pu être déterminé, à plus d'une trentaine d'années, et les tentatives faites pour découvrir l'existence possible de modèles plus récents ont échoué, vu que l'information pertinente appartenait au domaine privé. Il est d'ordinaire difficile de s'informer auprès des fabricants au-delà de ce que ces derniers révèlent dans les fiches techniques qu'ils publient.

La plupart des expérimentations faites sur les soupapes

de sûreté protégeant les wagons-citernes ont été menées par une équipe de l'Université du Maryland sous la direction de D.W. Sallet; elles ont permis d'étudier divers aspects de la question, comme les flux à l'intérieur de la soupape en conditions d'écoulement mono ou diphasique et les dépressions à l'intérieur de l'enceinte. Mais aucune des expérimentations du Dr. Sallet n'a pris en compte une soupape à levée libre du clapet. Celui-ci a toujours bridé à une hauteur de levée de consigne par une butée, ce qui n'a pas permis de connaître les caractéristiques dynamiques de la soupape, dont la connaissance est pourtant utile vu l'influence des pressions dynamiques sur la durée de vie de la soupape.

Bien qu'un certain nombre de programmes de modélisation informatique aient été trouvés dans le domaine, très peu ont fait l'objet d'une validation par comparaison des résultats théoriques à ceus tirés d'essais pratiques. Les rares cas recencés ont systématiquement abouti à de médiocres corrélations entre résultats théoriques et pratiques.

Autre faiblesse de ces modèles, ils ont toujours été associés à des sources modélisées par trop simplistes. Par source, on entend le système que la soupape est destinée à protéger: un wagon-citerne soumis à un feu extérieur, par exemple.

Les communications classées comme cas d'espèce concernaient surtout les programmes de maintenance et aucune ne renfermait de détails sur les soupapes. La recherche a montré que les programmes de maintenance et d'essais systématiques étaient rarement suivis. Là où il se fait des essais restreints, secteurs du nucléaire et des procédés chimiques, la recherche montre qu'il y a lieu soit d'améliorer la conception, le contrôle de qualité et les conditions d'utilisation des soupapes, soit de les repenser de fond en comble. L'expérience montre que les soupapes atteignent rarement les performances attendues. Souvent, les conditions d'écoulement à l'intérieur de la soupape conduisent à un phénomène appelé battement qui peut en endommager les organes internes et entraîner diverses anomalies: fuites et parfois même non-ouverture de la soupape aux pressions de consigne. Parfois aussi, la soupape est endommagée alors qu'elle n'a fonctionné que pour un instant très bref à la pression de début d'ouverture.

Très peu d'informations ont pu être glanées sur des accidents dus à la défaillance d'une soupape de sûreté. Dans la plupart des cas, l'incident à l'origine même de l'augmentation de la pression a été confondu avec la cause de l'incident, sans allusion aucune à la soupape ou à son mauvais fonctionnement. Même les données recueillies sur le comportement des soupapes de sûreté déclenchées, à l'occasion

d'essais sur des wagons-citernes montrent la médiocrité des performances. Dans les cas les plus graves, les défauts de fonctionnement de la soupape ont entraîné la rupture brutale de l'enceinte.

La recherche documentaire a permis de mettre en évidence les besoins suivants:

1) Expérimentations destinées à mettre en évidence les caractéristiques fonctionnelles des soupapes et à approfondir les forces dynamiques en jeu.

2) Les programmes d'entretien et de vérification dans les industries nucléaire et chimique sont insuffisants. Les procédures habituelles concernant le transport des produits dangereux doivent aussi être aussi examinées.

3) Validation des modèles de prévision assistés par ordinateur, associés à des sources modélisées appropriées, exemple le programme de modélisation des flux thermiques à l'intérieur des wagons-citernes, dans l'optique du perfectionnement des soupapes de sûreté.

4) Essais approfondis des soupapes prélevées sur une enceinte à pression accidentée afin de vérifier si elles ont bien fonctionné ou non. Les résultats de ces essais devraient faire partie intégrante du compte rendu de l'accident.

5) Élaboration d'une base de données sur les accidents mettant en cause des soupapes de sûreté.

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SECTION 1.0

INTRODUCTION

1.1 BACKGROUND

Tank cars carrying liquefied gases, such as propane, are equipped with a pressure relief valve (PRV) which automatically opens when the tank internal pressure rises above some predetermined value. If the tank car is exposed to fire impingement or any other high heat flux the tank internal pressure will rise, subsequently causing the relief valve to open and vent the tank contents into the atmosphere. On occasion, the relief valve malfunctions, resulting in slow responses, partial openings and in some cases even failure in the closed position. In the most serious cases, the malfunctions can lead to the uncontrollable rise of the tank's internal pressure. These increased pressures have been shown to cause catastrophic failures of tanks.

1.2 OBJECTIVE AND SCOPE

This report investigates the studies which have been conducted in the area of pressure relief valve technology. The original focus of the report was to conduct a literature search to find the state-of-the-art for PRVs which are presently being installed on railway tank cars, highway tankers, and stationary LPG storage vessels. When few papers were found which had concentrated on this particular topic, the authors continued the search by considering both the nuclear and chemical industries where similar technologies are found.

Such areas as experimental investigations of the fluid flow within PRVs, PRV computer modeling, and PRV maintenance requirements are addressed.

SECTION 2.0

LITERATURE SEARCH

2.1 OVERVIEW AND SOURCES OF INFORMATION

This literature review and search of pertinent data relating specifically to pressure relief valve systems for pressurized vessels was conducted by a manual search. Computer searches were not found to be useful for this study because they required that many subject headings be searched. This resulted in an overwhelming number of reports, of which few were pertinent.

Many of the references were compiled from the following data bases: Engineering Index, Applied Science and Technology Index, Transportation Research Board Publication Catalog, Highway Research Information Service Records, Railway Research and Engineering News, and U.S. Government Reports Annual Index.

After screening, that information deemed particularly relevant was placed into one of three categories: i) experimental studies; ii) modeling studies; or iii) case studies.

2.2 EXPERIMENTAL STUDIES

Information regarding maintenance requirements for railway tank cars was obtained from both the Association of American Railroads and the Canadian Transport Commission. Literature regarding maintenance programs for pressure relief devices for both the chemical and nuclear industries is also reviewed. The findings are discussed in detail in Section 4.1.2.

Section 4.1 also contains information regarding various experimental studies of pressure relief valves and rupture discs. D.W. Sallet of the University of Maryland has conducted the bulk of the research in the area of flow visualization and the effects of two-phase flow on the performance of relief valves presently being installed on railway tank cars. Other studies to determine pressure and flow-rate factors for valves being used in the nuclear industry have been conducted in the Soviet Union since the mid-1980's. These experimental investigations are discussed in Section 4.1.1 and summarized in Table 2.

2.3 MODELING STUDIES

The references selected for inclusion in this category have been further divided into four sections. The sections are as follows: 1) valve dynamic modeling; 2) source modeling; 3) flow modeling; and 4) effects of two-phase flow.

Many studies have been conducted in Canada, the United States, and the Soviet Union within the last decade which attempt to model relief valve characteristics. These models, and the sources which were employed in their modeling are presented in Sections 4.2.1 and 4.2.2, respectively. Section 4.2.2 also briefly discusses some of the more recently developed computer codes which model the effects of fire impingement on tank car performance. The majority of these models, which have been developed in Canada and the United States since the mid-1980's, have not coupled the tank cars with pressure relief valves.

During the last decade, researchers at the University of Maryland, Department of Mechanical Engineering, supported by the U.S. Department of Transportation, have conducted a detailed study of two-phase discharge from finite vessels. Other groups, such as the Design Institute for Emergency Relief Systems (DIERS), a consortium of 29 chemical companies within the American Institute of Chemical Engineers (AIChE), have also reviewed many of the existing two-phase flow models. The results of these studies are presented in Section 4.2.3.

Section 4.2.4 summarizes the pressure relief valve tests which have dealt with two-phase flow and its effects on valve performance. As discussed in Section 2.2, the majority of these tests have been conducted by D.W. Sallet at the University of Maryland.

2.4 CASE STUDIES

Very little information was found which related directly to accidents caused by relief valve failures. In the majority of cases, the incident which originally led to the rise in pressure was listed as the cause of the accident, and no mention was made of the relief valve performance.

Between the early 1970's and the present day, many small- and full-scale fire tests of tank cars have been conducted to investigate the performance of the vessels under accident conditions. Continued monitoring of internal pressure during testing has allowed the evaluation of pressure relief valve performance. Some of the more recent results are presented in Section 4.3.1.

SECTION 3.0

CURRENT PRESSURE RELIEF TECHNOLOGIES

3.1 PRESSURE RELIEF VALVES

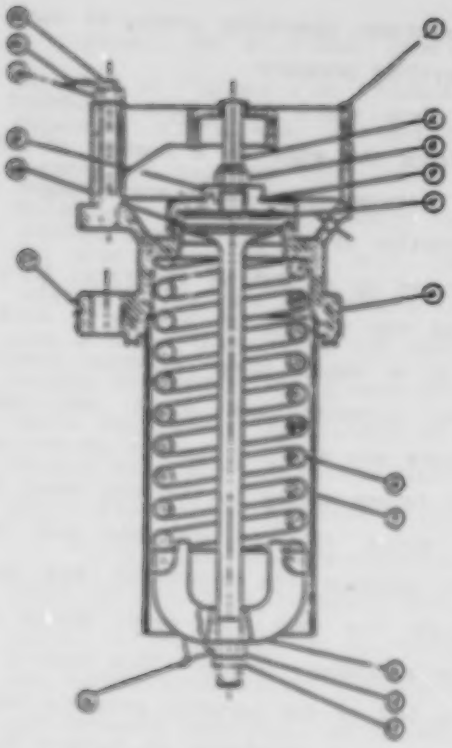
As near as can be determined, the Midland type relief valves currently installed on tank cars and highway tankers are based on designs more than 30 years old. Attempts to discover any current variations were unsuccessful because of the proprietary nature of the information. King (1981) commented on similar findings in an earlier literature search of pressure relief valves.

Figures 1 and 2 show a cross-section of a Midland type relief valve and its location on highway tanker cars. The valves are installed in a similar location on railway tank cars.

The operating principles of the PRV are straightforward. The spring-loaded pressure regulator is a continuous acting device. As the driving force (tank internal pressure) overcomes both the spring force and the force caused by the atmospheric pressure acting on the upper side of the disc, the valve starts to allow the tank-medium to escape. Further increases in tank pressure lift the valve disc from its seat and the flow rate increases. With some PRVs the flow is diverted downwards by a cup-shaped disc, and its direction is reversed. This action creates reaction forces that lift the disc further. The valve discharges to the atmosphere and its set pressure is regulated by a lock-nut located at the lower edge of the spring.

3.2 RUPTURE DISCS

Flat discs, conventional domed discs, pre-weakened discs, and reverse buckling discs are a few of the many types of rupture discs which have been developed in the past. Flat discs, the earliest and simplest of discs, are still commonly used. They are flat sheets of material weakened enough to burst before the system which they protect. The major disadvantage of these discs is that it is virtually impossible to predict their bursting pressure accurately. Conventional domed discs, as Mathews (1983) explains, are domed to form a portion of a sphere. The greater pressure is on their concave face. The behavior of these discs is analogous to that of a vessel under internal pressure: they burst when their ultimate tensile stress is exceeded. The problem with these discs



ITEM NO.	NO. REQ'D	PART NAME	MATERIAL
1	1	TOP GUIDE	STEEL WITH S.S. INSERT
2	1	STEM	STAINLESS STEEL
3	1	RETAINER	STEEL, CAD. PLATED
4	1	DISC	STEEL
5	1	SPRING	ALLOY ST., CAD. PLATE
6	1	FOLLOWER	STEEL, CAD. PLATED
7	1	SPRING GUIDE	STEEL
8	1	TOP LOCK NUT	STEEL, CAD. PLATED
9	1	WASHER	LAMINATED NEOPRENE
10	1	SEAT "O" RING	BUNA N 700
11	1	STEM "O" RING	BUNA N 700
12	1	SETT. ADJ. NUT	STEEL, CAD. PLATED
13	1	SETT. LOCK NUT	STEEL, CAD. PLATED
14	4	STUD	ALLOY STEEL
15	4	STUD NUT	STEEL
16	1	SEAL	LEAD WITH S.S. WIRE
17	1	NAME PLATE	STAINLESS STEEL

Figure 1: Cross-section of a Midland Type PRV

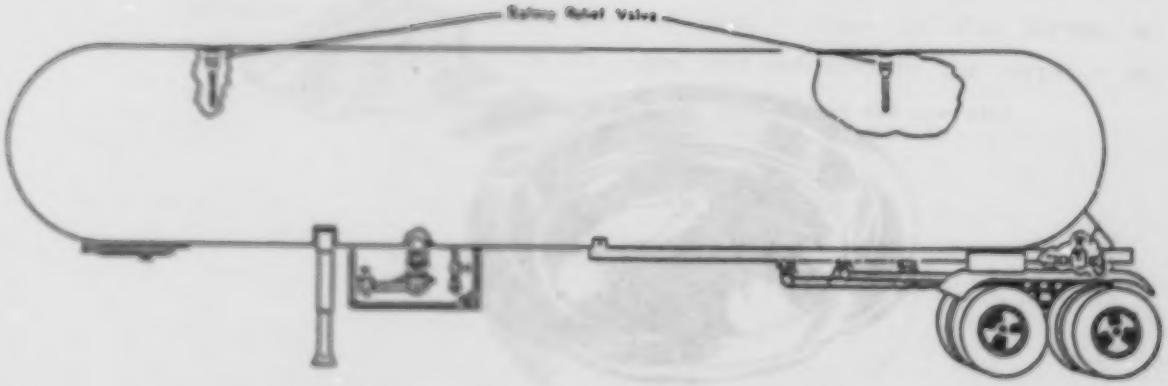


Figure 2: Typical PRV Locations on Highway Tankers

is that, because of susceptibility to creep, operating pressures must be limited to about 70% of their rated bursting pressure.

Pre-weakened discs are thick discs which have been weakened by cutting radial slits, for example, from the edge to a number of small holes near their centre (Figure 3). Because these discs are stronger and stiffer than conventional discs, distortion creep and fatigue become less of a problem and they can be operated up to 80% of the rated bursting pressure. Mathews (1983) describes the actions of a reverse buckling disc as being analogous to that of a vessel under external pressure. Below the bursting pressure the stresses are small and mainly compressive. At the buckling pressure any slight distortion of the shape displaces the line of action of the forces acting in the disc away from the centreline of the material, giving rise to moments and hence increased stresses which in turn increase the distortion and so on towards sudden collapse. Eventually, the disc flips over, where it is cut open by a sharp knife or other equivalent device. This type of disc permits working pressures of up to 90% of the bursting pressure and, as Zanetti et al (1984) indicates, they have a resistance to fatigue that is two hundred times greater than that of standard units. The different disc types along with a more detailed explanation of the advantages and disadvantages of each is presented in Section 4.1.3.

Figure 3: Pre-weakened Rupture Disc.
Adopted from Mathews (1983)

Zanetti et al (1984) briefly describes some of the most recent advances in rupture disc technology. BS & B Safety Systems' in Tulsa, Oklahoma offers what is said to be the first reverse-buckling device capable of handling both liquid and gas. Previously, reverse-buckling discs have not been successful in liquid service since the liquid deforms the disc too gradually, with insufficient "snap" to provoke a sudden bursting. The new disc has a C-shaped score line (groove) cut into it, plus a hinge that forces the device to crack along the C before it can become inverted. The groove ranges in depth from 50% to 90% of the disc's thickness. As Zanetti et al (1984) indicates, this new disc is finding application in polymerization equipment and railroad tank cars, which often contain mixtures of liquids and gases.

BS & B has also developed another new disc known as the MRB, in which the disc and its holder are machined from a single piece of metal instead of separately, as is usually done. The unit, said to last up to two million cycles at 0 - 90% pressure variation, is particularly good in high-pressure operation, up to 125,000 psig. It is being used in pumps for cement, drilling mud, and slurry.

Another device available, as Zanetti et al (1984) explains, is a burst-disc indicator that uses an infra-red photoelectric sensing system to detect breakage. The device, manufactured by Fike, consists of a control unit mounted externally on the piping, a fibre-optic cable, and a reflective target fastened inside the pipe. When the disc bursts, a light beam - originating in the cable, and reflected by the target - is interrupted. This closes an alarm circuit that warns the operator.

SECTION 4.0

DISCUSSION

SECTION 4.1

EXPERIMENTAL STUDIES

4.1.1 VALVE TESTS

Several experimental investigations have been conducted on pressure relief valves (PRV's) in the last decade. Specific areas of interest have included valve performance during vibratory excitation, valve performance using various flow regimes, flow visualization and determination of valve pressure and flow-rate coefficients. These tests are summarized in Table 2, and will be discussed in greater detail here.

Bordelon et al (1979) studied the vibration effects on the functioning of a PRV by conducting a series of shaker table tests on a 6" pressure relief valve. The spring, spindle, disc, and disc holder in a relief valve are "free" bodies and are held together only by the vertical compressive spring force. Lateral movement at both the top and bottom of the valve are restricted by the upper support structure and the base assembly respectively. During a dynamic disturbance, the "free" bodies may rock or move laterally depending on the relationship of spring load, component flexibility and levels of acceleration.

The test data of Bordelon et al (1979) indicated that when pressure relief valves were subjected to vibration, the force balances on the valve components were different from those observed under non-vibratory conditions. If vibration was sufficiently severe, the actual opening pressure of the PRV was different from the opening pressure expected under non-vibratory conditions. Opening pressure was lower when the PRV was subjected to increasingly higher horizontal accelerations. When the valve was subjected to vertical accelerations, the opening pressure changed in a random fashion. The latter observation was explained by the fact that vertical base excitation only slightly disturbs the spring force system in the valve. The effect is too small to cause a consistent drop in valve opening pressure.

Coffman et al (1980) investigated the flow-induced sonic vibration in boiler safety valve nozzles which had been observed to cause premature valve

wear and failure on large boilers at Oklahoma Gas and Electric Company. The report investigated possible causes of vibration and possible reasons for the success of a tapered shape reducer in eliminating the vibrations. The limited design guidance available for safety valve placement was discussed and the report concluded by presenting the results of a safety valve vibration experience survey. Coffman et al (1980) suggested a two-parameter guideline for safety valve placement, involving steam velocity and valve location, the results of which are presented in Figure 4.

In the late 1970's and early 1980's D.W. Sallet at the University of Maryland conducted several experiments on PRVs under contracts from both the Electric Power Research Institute (EPRI) and the U.S. Department of Transportation, Federal Railroad Administration. Most of Sallet's efforts were towards understanding the flow within PRVs under different medium flow, and the effects of the different flow regimes on valve performance. Sallet (1978a), (1979a), and (1979b) developed and reviewed methods of sizing the PRVs which are installed on tanks containing compressed liquefied gases. These and other discussions on PRV sizing, such as much of the content of a study conducted by a special ad hoc committee of the AAR Tank Car Committee (1983), fall outside of the scope of this literature search, and thus are not discussed further in the present report.

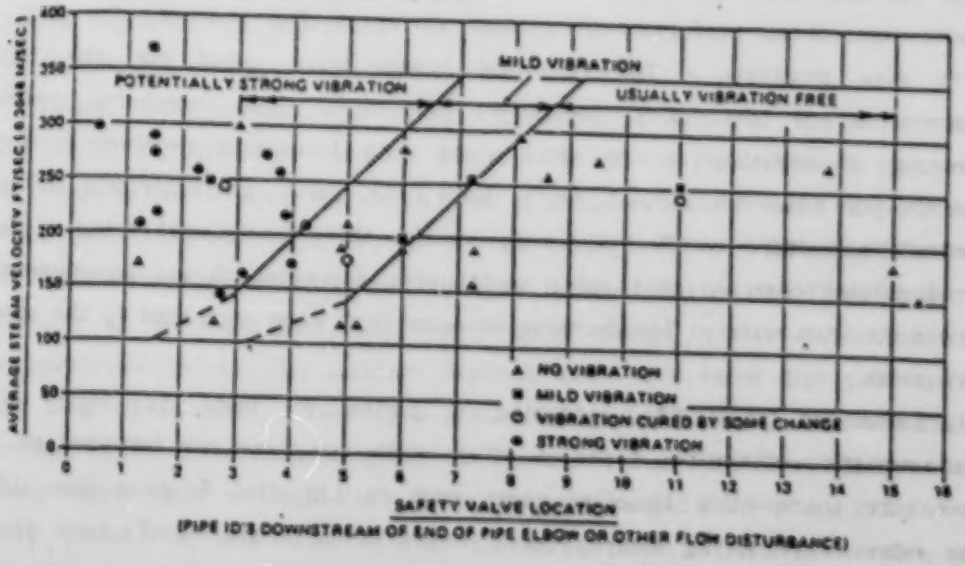


Figure 4: Plot of Steam Velocity/Valve Location Vibration Experience

Adopted from Coffman et al (1980)

Sallet et al (1981a) studied the flow and pressure distribution inside a typical PRV as well as other parameters such as valve geometry, extent of opening, etc., which affect the performance and relief capacity of the valve. The types of tests which were carried out included 1) flow visualization; 2) choked air flow tests; 3) incompressible flow tests with water; and 4) two-phase choked flow tests with Freon and water. Flow visualization tests were performed on a double scale, two-dimensional plexiglas half-model of the PRV to determine the flow pattern in the valve. The setup for the visualization tests consisted of a closed, steady flow loop allowing water to flow through the valve model. Air was injected into the flow through hypodermic needles and was obtained from a compressed air supply using flow regulating and measuring devices. Optical measurements of the direction and magnitude of the water velocity in the valve model were made with a Laser-Doppler-Anemometer. High velocity gradients and changes in flow direction indicated that flow separation occurred within the valve. A large vortex at the exit of the valve was noted, and from photographs it appeared that the minimum valve cross-section was even further reduced due to flow separation.

The choked air flow tests used two- and three-dimensional models of the PRV. In both models, the disc and adjusting ring were movable. A series of twelve tests were performed using four different disc positions for each of three ring positions. The following trends were noted for the twelve experiments: 1) the valve coefficient decreases with increasing receiver pressure; 2) pressure in the valve rises with increasing receiver pressure; and 3) the two-dimensional tests show the same characteristic internal pressure and valve coefficient trends as the three-dimensional tests. Sallet et al (1981a) observed that valve coefficients decreased during incompressible flow tests with water. The decrease exceeded 30% when compared to the choked flow tests.

Sallet et al (1982) conducted a series of tests to verify flow characteristics within two types of PRVs which are commonly installed on rail tank cars transporting liquefied gases such as Liquefied Propane Gas (LPG). The valves were spring loaded poppet valves of different manufacture and of different design. To determine valve coefficients a series of laboratory tests were designed to simulate the different flow regimes and to realize the

flow patterns which may occur at any possible service condition of the valves. Flow visualization tests were carried out for liquid flows, vapour flows, and two-phase flows. A survey of the different test series is given in Table 1, and an example of the velocity surveys is presented in Figure 5. Figure 5 shows the outline of the model of one of the PRVs in the fully open position. It also shows velocity profiles of water taken in the mid-plane of the model at 6 stations, namely, in the throat, at the edge of the seat, at one radial and three axial sections in the outlet annulus of the valve for two conditions. The two conditions presented here are 1) pure water flow at 1.1 ℓ/sec; and 2) two-phase flow at 0.7 ℓ/sec of water and 0.4 ℓ/min of nitrogen gas, with the volume of the nitrogen gas being referred to standard atmospheric conditions.

Sallet et al (1982) observed during these tests that entrainment of liquid droplets decreased the mass rate of vapour flow, yet increased the combined mass rate of flow; that an increase in droplet size increased the combined mass rate of flow; and that for a given liquid flow rate the droplet size very strongly influenced the pure vapour flow rate, with finer spray decreasing the vapour flow rates. Sallet et al (1982) also observed the following: 1) valve coefficients remained nearly the same when pure liquid flow changed to very low quality two-phase flow; and 2) entrainment of liquid droplets significantly decreased the vapour flow rates, the decrease being most pronounced during choked flow.

Sallet (1984) conducted a literature review of some of the most recent work on PRVs. His paper included a discussion of the fluid dynamic phenomena that lead to valve disc vibrations and a presentation of some experimental results describing one- and two-phase flow through PRVs. Sallet (1984) found that the minimum cross-sectional flow area is dependent on the flow separation within or at the valve exit, and that it could change in size and location for any particular valve. Its location depends upon both valve disc position and valve ring adjustments. Test data presented by Sallet (1984) showed that a decrease in valve coefficient could be expected when the flow changes from choked vapour flow to liquid flow. The decrease ranged from 16 to 27% for the valves tested. It was observed that the streamlined pattern in a valve does not change abruptly when the flow regime changes from a pure liquid to a gas/liquid mixture at very low void fractions and low velocities.

Test Series	Valve	Medium	Flow Regime	Test Purpose	Valve Location & Discharge Conditions
I	full scale; actual valves (A) and (B)	air	choked flow	simulation of choked vapor flow; determination of valve coefficient for choked flow; study of valve performance during initial valve lift period	valve is in vapor space; droplet entrainment is negligible; valve operates at design conditions
II	full scale; actual valves (A) and (B)	air	low velocity; (incompressible flow) gas flow	simulation of low velocity vapor flow; determination of valve coefficient for low velocity vapor flow; check on liquid flow valve coefficient obtained from Test Series III	valve is mounted for sizing in typical volume flow measuring apparatus
III	full scale; actual valves (A) and (B)	water	low velocity; liquid flow	simulation of liquified gas flow in the absence of cavitation and flashing; determination of valve coefficient for liquid flow; check valve modeling of Test Series V	tank car has overturned and valve is in liquid space; valve discharge conditions are such that liquid flow occurs through the valve; the fluid flashes outside the vessel
IV	full scale; actual valve (A)	water & air	choked flow flow with liquid droplet entrainment	simulation of choked vapor flow with droplet entrainment; estimation of entrainment effects on vapor discharge rates	valve is in vapor space but near a boiling liquid surface; vapor flow with liquid entrainment takes place
V	full scale; 2 dim. model of valve (A)	water	liquid flow	study of flow pattern in valve during liquid flow	same as Test Series III
		water with N_2 bubbles	low quality two-phase flow	comparison of two-phase flow pattern with liquid flow pattern; estimation of valve coefficient for low quality two-phase flow	valve is initially in vapor space but experiences two-phase flow due to violent boiling(heat input due to fire); or, tank car has overturned and valve is in liquid space and two-phase flow occurs

Table 1: Survey of Sallet's Test Series

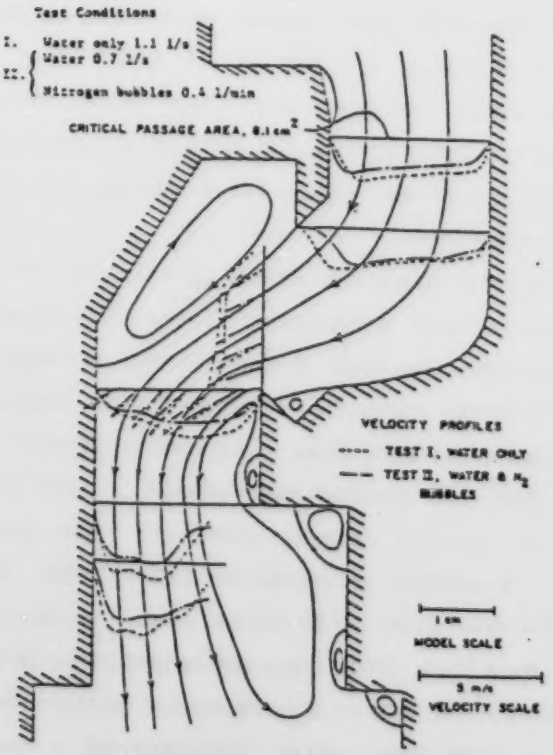


Figure 5: Measured Velocity Profiles and Streamline Pattern of Flow Through the Two-Dimensional Valve Model.

Table 1 and Figure 5 Adopted from Sallet et al (1982)

Sallet (1984) described the fluid dynamic phenomena that may lead to valve disc vibrations. He suggested that the most important causes of valve disc vibrations in PRVs are the type of coupling between flow forces acting on the valve disc trying to increase valve disc lift, and the spring force trying to restrain valve lift. Other causes include the dependence of valve lift on total mass flow rate, and the dependence of total flow rate on the flow losses. He explained that flow losses were generally higher in liquid flows than in gas flow, and that the losses were higher, again, in two-phase flows. Sallet (1984) also suggested that the interaction between shock waves and flow separation could cause oscillating flow fields in a safety valve.

Sallet et al (1985) conducted tests on PRVs using saturated water flow as the medium. It was observed that the volume flow rate was greatly decreased when the flow medium was saturated liquid as opposed to subcooled liquid. At a vessel pressure of 100 psia the decrease in flow rate for saturated water amounted to 60% of the flowrate observed during subcooled liquid conditions. When saturated or nearly saturated fluid flowed through the PRV, the valve disc experienced large amplitude vibrations. These vibrations were not present when the flow regime was a highly subcooled medium. The presence of vibrations led Sallet et al (1985) to conclude that the cures for valve disc vibrations must be addressed.

Popov et al (1982) studied the effects of the flow conditions in valve slots and of the flow structure on the stability of PRVs using flow visualization techniques. He explained that vibrations observed within valves were caused by the unsteady state of the flow in the valve slot owing to the periodic detachment of eddies when there was a change in the structure of the flow caused by cavitation. He observed that for Reynold's Numbers of less than 6.1×10^3 closed pockets were present within the valve. As the flow rate increased the pockets got longer and immobile eddy regions appeared in the output chamber. When the Reynold's Number exceeded 6.1×10^3 the flow changed with the result that two fluid jets appeared. The jets were inclined at approximately the same angle both to the axis of the valve and to the walls of the valve slot. With further increases in the flow rate the inclination of the jets from the valve axis became greater and fluid was expelled from the pockets of the output chamber. With increases in back pressure, eddies began to appear in the pockets. The formation and removal of the eddies occurred

periodically. A change in the angle of the flow was observed when the eddies were being removed. After removal of the eddies, the jets were suddenly inclined at a greater angle.

Analysis of the test results of Popov et al (1982) suggested that one of the causes of valve vibration was the periodic change in flow structure. By developing a mathematical model of the flow within the valve it was concluded that three conditions had to be satisfied if the observed type of vibration was to be avoided. The three conditions were: 1) that loading conditions must not change the natural frequency of the valve; 2) that back pressure be absent; and 3) that the period of eddy formation must be less than half of the valve's natural vibration.

Many experimental studies of the flow rate and force characteristics of PRVs for single- and two-phase working media have been conducted in the Soviet Union. The experiments of Kolenko et al (1984) dealt with the dependence of flow-rate factor of the medium in safety valves on the height of lift of the valve above the seat. For all valves examined the flow-rate factors increased monotonically with increasing valve-lift height. This was true for both sub- and supercritical flow. Flow rate factors with supercritical flow were observed to be lower than those obtained with subcritical flow. After completion of experimentation, Kolenko et al (1984) developed empirical relationships to estimate the flow-rate factor under both sub- and supercritical conditions. The factors predicted using these equations approximated experimental data satisfactorily.

Borzov et al (1984) experimentally obtained functions of nondimensional flow and pressure coefficient versus valve lift height for high-lift, angle-type safety valves using various flow-path configurations. When superheated steam was used as the working medium, the flow was found to be supercritical when (atmospheric pressure) / (valve inlet pressure) was less than 0.3. Lifting forces were increased with the addition of deflectors on the disk because of the increased disk area and the reaction of the flow being deflected by the disk. Since the main component of the force exerted on a disk with deflectors was caused by a drop in pressure, disk forces were constant over the entire range of disk travel. Borzov et al (1984) also presented a semi-empirical model of a safety valve actuation process. The model was not, however, compared with any experimental results.

Borzov et al (1986a) conducted further experiments to determine PRV characteristics when two-phase flow was present. The presence of moisture in the heat carrier steam had a significant influence on the main dynamic characteristics of the PRV. It was observed that for valves with conical deflectors the presence of moisture in the stream had practically no influence on the stress exerted by the stream on the slide valve when the lifting height of the slide valve was low. As the lifting height increased however, the stress exerted by the two-phase stream on the slide valve was reduced by 30% in comparison with single-phase streams. It was found that the passage of a two-phase stream through the seat-slide valve section was characterized by sharp separation of the phases and the occurrence of a stratified flow regime. Pulsations in the forces acting when the lifting height of the slide valve was low, noted by Borzov et al (1984), were again noted and the conclusion was drawn that this type of safety valve was not appropriate under these conditions. It is hoped that current research being conducted at the Department of Mechanical Engineering, Queen's University, will determine whether the PRVs installed on rail tank cars produce similar results as those obtained by Borzov et al (1986a).

Borzov et al (1986b) continued his study of the power characteristics (analysis of forces acting) of PRVs by conducting tests with and without seal movement. It was discovered that the characteristics observed under both scenarios were similar. Borzov et al (1986b) concluded that the use of power characteristics obtained in the stationary conditions was justified when designing nonstationary operational processes of direct-acting PRVs.

TITLE	AUTHORS	RESEARCH INTERESTS	FINDINGS	RECOMMENDED AREAS OF STUDY
Vibration Effects on Pressure Relief Valve Function 1979	Bordelon, T.R. Lai, T.S.	- to study vibration effects on the functioning of a PRV by conducting a series of shaker table tests.	- opening pressure drops when the valve is subjected to a vibratory excitation. - horizontal base excitation produces larger effects than does vertical base excitation. - to reduce vibration: adjust valve under expected vibratory conditions or decouple structural dynamic effect between valve and supporting elements.	- no comment
Failure of Safety Valves Due to Flow-Induced Vibration 1980	Coffman, J.T. Bernstein, M.D.	- to develop a 2-parameter guideline for SV placement, involving steam velocity and location of SV with respect to upstream bends. - this is accomplished by conducting a survey of SV installations.	- a graph of valve location versus steam velocity is presented. The plot is divided into 3 sections: potentially strong vibrations, mild vibrations, and usually vibration free. - by finding where on this plot a proposed SV installation falls a designer can assess the likelihood of the occurrence of vibrations.	- more plots are needed to extend & confirm results obtained thus far.
An Experimental Investigation of the Internal Pressure and Flow Fields in a Safety Valve 1981	Sallet, D.W. Mastelli, V. Knight, R.W. Palmer, M.E.	- to study flow and pressure distributions inside a typical safety valve (with fixed valve disc). - types of tests carried out: flow visualization, choked air flow, incompressible flow tests with water, two-phase choked flow with Freon & water.	- the flow pattern in the SV is complex and has several separation points. - separation decreases the effective minimum flow cross section and decreases mass flow rate. - increased upstream reservoir pressure decreases the amount of flow separation.	- additional work dealing with various aspects of 2-phase flow: the question of thermal equilibrium during flow through the valve and the problem of flow separation. - future tests should include valve dynamics, ie. valve disc free to vibrate.
Effects of the Flow Conditions in the Valve Inlet and of the Flow Structure on the Stability of a Safety Valve 1982	Popov, D.N. Ofrouzkh, F.V.	- to examine the effect of the unsteady state of fluid flow caused by cavitation on SV characteristics.	- vibrations of the valve can occur if there are unsteady eddy zones in the output chamber. - to avoid vibration: a) loading conditions must not change the natural frequency of the valve; b) back pressure should be absent; c) the period of eddy formation must be less than half of the valve's natural frequency.	- no comment
The Flow of Liquids and Gases Through Rail Tank Car Safety Valves 1982	Sallet, D.W. Weeks, J.S. Guhler, K.	- to perform a series of tests to verify flow characteristics of PRV's being frequently installed on rail tank cars. - flow regimes studied: liquid, vapour and two-phase flows.	- valve coefficients for low velocity vapour and liquid flow are 30% less than choked vapour flow coefficients. - valve coefficients are nearly equal for liquid and low quality two-phase flows. - entrainment of liquid droplets decreases mass rate of vapour flow yet increases combined mass rate of flow. - increased droplet sizes increase combined flow rate. - for a given liquid flow rate, decreasing the droplet size decreases the vapour flow rates.	- no comment

TABLE 1: Survey of Experimental PRV Tests

TITLE	AUTHORS	RESEARCH INTERESTS	FINDINGS	RECOMMENDED AREAS OF STUDY
Determination of the Flow-Rate Factor of the Medium in High-Lift Safety Valves 1960	Antenko, S.E. Dedkov, A.S. Malyshin, G.P. Semenshov, Yu. I.	- to determine the dependence of the flow-rate factor of the medium in high-lift gas on the height of lift of the valve.	- flow-rate factors of all examined valves increase monotonically with an increase in valve lift height both for sub- and supercritical flow of medium. - flow-rate factor in supercritical flow is lower than in subcritical flow. - authors develope eqns. to estimate the flow-rate factor which approximate satisfactorily experimental data.	- no comment
Study of the Characteristics of Angle-Type Safety Valves 1980	Berzov, E.A. Grishin, Yu. M. Kozlov, V.M.	- to obtain functions of flow-rate and pressure coefficients versus valve lift height for high lift, angle-type safety valves	- working medium flow is found to be supercritical when (outlet pressure)/(pressure at valve inlet) < 0.5 - deflector on disk increases lifting force because of increased disc area and reaction of the flow deflected by the disc. - for a disc with a deflector, force is constant over entire range of disc travel since the force is mainly determined by pressure drop. - using different disc configurations permits effective control of power characteristics.	- no comment
Flow Capacity and Response of Safety Relief Valves to Saturated Water Flow 1985	Sallet, D.W. Somers, G.W.	- to study the effect of exit to nozzle area ratio on the flow capacity and on the stability of a valve. - to study mass flow rates of saturated liquid and low quality liquid-vapor mixtures through valves.	- the volume flow rate of saturated water is much lower than that found for subcooled water - for highly compressed liquid and saturated liquid restricting valve outlet does not decrease volume flow rate unless exit to nozzle area is 5 or less. - bellows significantly increase the volume flow rate of saturated liquid for valves of exit to nozzle area ratios less than 4. - no valve disc vibration occurred with a highly subcooled medium. - the valve discs experience large amplitude vibrations when saturated fluid was flowing. - severe valve vibrations occurred for the valve with bellows and with exit to nozzle area ratios greater than 10.	- experimental results point towards the need to address cures for valve disc vibration.
Hydrodynamic Characteristics of Safety Valves for Two-Phase Heat Carriers. 1986	Berzov, E.A. Grishin, Yu. M. Kozlov, V.M.	- a study of the flow rate and force characteristics of valves subjected to two-phase streams.	- for valves with conical deflections & with low lifting heights of the slide valve, the pressure of moisture in the stream has practically no influence on the stress exerted by the stream on the slide valve. - at large lifting heights stress exerted by the two-phase stream on the slide valve is reduced by 20% in comparison with the 1-phase stream.	- no comment
Effect of the Rate of Seal Movement in the Power Characteristics of the Safety Valve. 1986	Berzov, E.A. Kozlov, V.M. Kurnatsov, O.V.	- to determine the power characteristics of a valve with and without seal movement.	- the use of power characteristics obtained with stationary conditions is appropriate when designing nonstationary operational processes for valves.	- no comment

TABLE 1: Survey of Experimental PRV Tests (Continued)

4.1.2 MAINTENANCE/PERFORMANCE-TESTING PROGRAMS

As Woolfolk et al (1984) explained, all of the effort spent sizing and engineering a system to give adequate protection is of little value if it is not followed by an adequate inspection and maintenance program using established realistic inspection frequencies. This section begins by reviewing the standards and recommended practices of both the Association of American Railroads and the Canadian Transport Commission. Little information was uncovered which dealt with actual programs being conducted by such industries as rail tank car manufacturers. Instead, some literature was found relating to inspection programs being conducted by both nuclear and chemical industries. This section describes these testing programs in detail and provides a summary in tabular form (Table 3). Table 4 is a conglomeration of information describing valve failure modes and maintenance tips. The distinction between the two tables is that reports presented in Table 4 represent the experiences of the authors. Actual tests are not described.

Detailed requirements for safety relief devices are contained in the Association of American Railroad's Manual of Standards and Recommended Practices, Section C - Part III, Specifications for Tank Cars (Specification M-1002). Other such specifications include those given by the Canadian Transport Commission's "Regulations for the Transportation of Dangerous Commoditites by Rail".

The standards and specifications state acceptable materials and operating characteristics (such as allowable variation in set pressure) for the PRVs installed on rail tank cars. The codes discuss the stamping of the valves and indicate that such items as manufacturer, manufacturer's design number, set pressure, official flow capacity and date of manufacture must be clearly marked on each valve. Codes require that repair work on safety relief devices involving machining, grinding, welding or other alterations or modifications only be performed by the valve manufacturer or by the car owner or user with the valve manufacturer's permission.

A rigorous testing procedure is required for new designs of PRVs. The new design must be tested for a minimum period of two years and an average of 10,000 loaded miles or an average of twenty loaded trips or load/unload cycles. Difficulties experienced during the service trial (such as leakage, breakage, or performance) must be reported along with design revisions. After

the testing period, the valves must be thoroughly examined for defects before final certification is given.

Hunter (1982) discussed the results of an EG & G Idaho, Inc.'s test program which demonstrated the adequacy of safety/relief-valve systems under anticipated reactor operating and accident conditions. Ten designs of relief valves were tested under PWR operating steam conditions. The results revealed many problem areas. Problems identified included incorrect trim installation within the valve, plug/cage material incompatibility, and improper dimensional tolerances on valve stems.

Between 1980 and 1982 the Electric Power Research Institute conducted a safety and relief valve test program at three sites. As O'Keefe (1984a) explained, fluid conditions during testing included steam alone, steam-to-water transition, and water at several temperatures. Very discouraging results were observed. Such defects as a bent stems, failed welds, defective machining in bellows assemblies, and washout of cage/body gaskets in relief valves were encountered. Operating deficiencies such as sticking of internal moving parts, causing partial lift, closure delays, and failure of relief valves to close fully were observed. With safety valves, water caused chatter or flutter, and any amount of this resulted in internal damage. O'Keefe (1984a) concluded by suggesting that his tests indicated that either many minor improvements in design, quality control, and application were needed, or else that fundamental rethinking was called for in safety and relief valves.

Woolfolk et al (1984), of PPG Industries, Inc., Lake Charles, La., described the ongoing safety relief valve testing program which operates within that company. The program employs two full-time, factory-trained mechanics who are responsible for maintaining the 2800-plus safety relief valves on site. Tests are conducted using a clampdown table with a 1/4" tubing from an 85 ℓ pressure cylinder. Test frequency for storage vessels with no heat source is two years.

Woolfolk et al (1984) identified the need for the removal and testing of new safety valves during the pre-operative stage for all new installations. On two projects, PPG found that more than 30% of new safety relief valves did not function within ASME Code (2) tolerances. It was realized that after manufacturers had set and sealed safety valves to their satisfaction, they had

very limited control over the handling and storage of the safety valves. Such factors as rough handling, mud, rain, sandblasting, painting, and chemical exposure from adjacent plants were identified as causes of safety valve malfunctioning. Some of the actual problem areas identified included minute leakage through the seat, chattering, and corrosion of parts. A more detailed description of the findings is presented in Table 3.

In 1980, 1972 boiler and pressure vessel accidents, 108 injuries, and 22 deaths were reported to the National Board of Boiler and Pressure Vessel Inspectors. The overwhelming number of incidents occurring each year prompted the National Board to conduct a survey among the chief boiler and pressure vessel inspectors of the various jurisdictions - for the states and large cities of the United States and for the provinces of Canada. The objective of the survey was to identify the most common unsafe conditions associated with accidents. The distinctly unique group of people for whom the survey was created were chosen because of their vast practical experience and knowledge of boiler and pressure vessel problems.

As Jagger (1984) explains, the survey essentially asked the question: "Based on your experience, how likely are you to find each of the following conditions during a typical accident investigation?" Frequency was tabulated as "frequent", "sometimes", or "rare". To rank the responses, the frequencies were weighed. Pertinent results of the survey are presented in Figures 6 and 7. Of the nine general problem areas identified, "poor maintenance and testing of controls" ranked first in frequency. Within this general category safety valve failures ranked highest and were observed to occur "quite frequently." Results indicated that regular testing programs for safety devices were not followed. The needs for the formulation and adoption of good construction and installation codes and standards, as well as regular third party inspections by the National Board were identified.

Esso Chemicals in Sarnia, Ontario, test their three or four thousand safety valves every 12 to 24 months. Valves used on corrosive or gummy substances are tested twice annually. The testing program, as described by Pylyshyn (1986), involves pulling the valves off, stripping them down and rebuilding them. A schedule is prepared one year in advance, indicating when each valve will be sent to their on-site machine shop. One month before the repair date, valve specifications are sent to the repair shop, to make sure

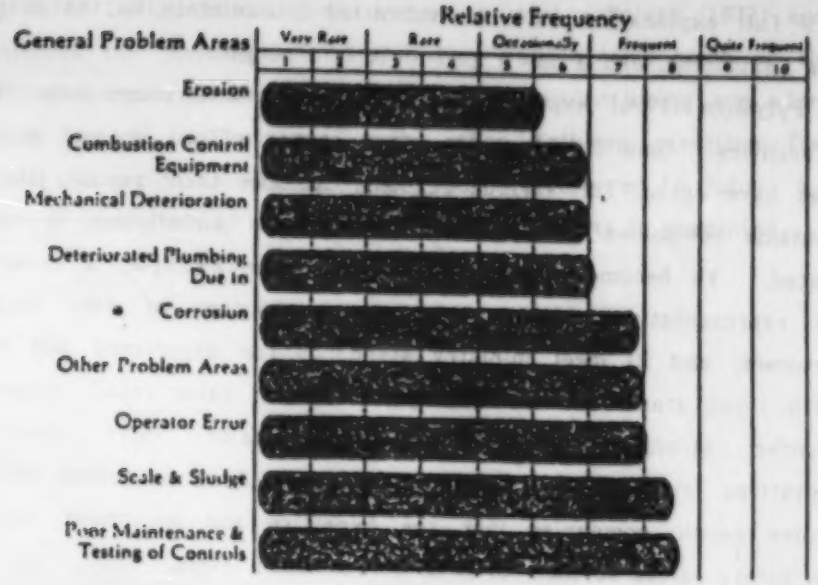


Figure 6: National Board Survey Results
General Problem Areas

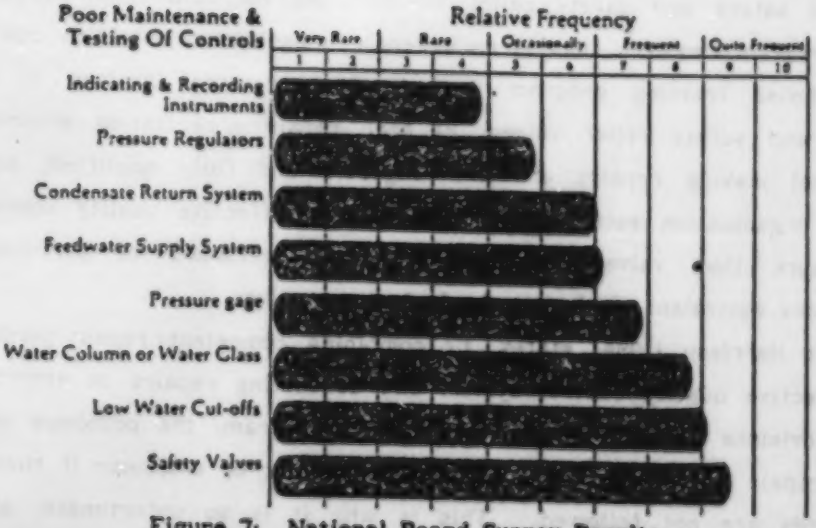


Figure 7: National Board Survey Results
Poor Maintenance & Testing of Controls

Figures 6 and 7 Adopted from Jagger (1984)

the correct parts are in stock at maintenance time. If a scheduled check is missed, a full explanation must be submitted immediately to the plant's Safe Operating Committee, and a new repair date is arranged.

As Pylyshyn (1986) explains, not all safety valve users have their own repair facilities. Smaller companies often cannot afford it, and many larger companies have opted, for various reasons, to take their repair jobs outside. These outside companies must be certified by the jurisdiction in which they are located. To become certified in Ontario, for example, a company must 1) have representatives attend the training courses of two major valve manufacturers; and 2) meet ministry standards for experience and equipment. Even with these standards, the quality of Ontario valve repair companies has been under strong criticism from the major valve manufacturers. Representatives from Dresser, Crosbie and Farris have expressed concern that many valve repair companies lack the expertise and equipment necessary to maintain safety valves at their original performance levels.

In January 1977, the National Board Executive Committee approved procedures for the issuance of a National Board Certificate of Authorization for the use of a valve repair "VR" stamp. The certification program was introduced to address the need for adequate repairs to ASME/National Board stamped safety and safety relief valves. As Harrison (1984) explained, the "VR" program requires that 1) each repair organization establish and document an in-house training program for all those persons involved in repair of safety and safety relief valves; 2) each repair organization ensure that its personnel making repairs are knowledgeable and fully qualified; and 3) the repair organization establish and maintain an effective quality control system to ensure that valves repaired have been returned to performance and conditions equivalent to the standards for new valves.

As Harrison (1984) states, by combining competent repair personnel with an effective quality control system and conducting repairs in accordance with the provisions of the National Board "VR" program, the problems with safety valve repair can be overcome. Problems will not be overcome if these exacting standards are not followed. This is why it is so unfortunate, as Pylyshyn (1986) points out, that very few Canadian repair shops follow these standards. In 1986, the number of repair shops in Canada following the standards was a mere two.

Table 4, as indicated previously is a conglomeration of information regarding valve failure modes and maintenance tips. Bright (1972) and Garvin (1981) discuss the jamming of relief valves and identify some of the major causes. Chattering, and other causes of valve malfunctioning are addressed in the various papers, as are the many other failures resulting from poor maintenance and design techniques. The reader is referred to Table 4 where a more detailed description of the findings of these reports is presented.

TITLE	AUTHORS	TEST PROGRAM AND INDUSTRY	TESTING PERIOD	MAJOR FINDINGS	COMMENTS AND RECOMMENDATIONS
Safety/Relief Valve Test Program 1982	Hunter, J.A.	Tests conducted by E.G.&G. Idaho, Inc. to study performance of safety/relief valve systems. Nuclear	1982	-test acceptance requirements stipulated that valves open and close on demand. -problems encountered include: plug/cage galling because of incorrect trim installation yielding unacceptably small tolerances; galling because of plug/cage material incompatibility; valve stems with improper dimensional tolerances.	No comment.
Performance Test on Valves Highlights Rigors and Dilemmas of Nuclear Service January, 1984	O'Keefe, W.	Electric Power Research Institute (EPRI) Safety and Relief Valve Test Program Nuclear	1980 - 1982	-such defects as bent stems, failed welds and defective machining in bellows assemblies, and washout of cage/body gaskets were encountered. -operating deficiencies included sticking internal moving parts, causing partial lift, closure delays, and failure of relief valves to close fully. -blowdown was often far above nominal design figures, even after repeated adjustment. -the presence of water caused chattering or flutter, and any appreciable amount of this resulted in internal damage.	Either many minor improvements in design, quality control, and application are needed, or else fundamental rethinking is called for in safety and relief valves.
Process Safety Relief Valve Testing March, 1984	Woolfolk, W. Sanders, R.	Safety relief valve testing program, PPG Industries. Chemical	1974-present	-minute leakage through the seat will carry solids out of the valve and into close tolerance guide clearances, it also causes deposits in the seating area itself. This may restrict lift. -chattering can drastically reduce capacity of the valve and can result in destruction of the safety valves. -many parts are found to be corroded or incorrectly machined, causing unbalanced flow across the seat and resulting in binding in the guiding area. Corrosion in guiding areas is the most common cause of late pops. -defective parts restrict valve seats from reaching full lift at design overpressure, cause misalignment that could result in leakage, or cause the seat to remain in the closed position regardless of vessel pressure. -springs weakened during service lead to early simmer and excessive blowdown. In many cases early simmers, such as 20-30% of the actual popping pressure, can not be corrected. -on two multimillion dollar projects in the early 1980s it was found that more than 30% of all new safety relief valves would not function within ASME Code (2) tolerances. -even brief operation at the set point can damage a safety valve.	Regular inspections of all safety valves is a necessity to ensure proper operation.

TABLE 3: MAINTENANCE PROGRAMS (CONTINUED)

TITLE	AUTHORS	TEST PROGRAM AND INDUSTRY	TESTING PERIOD	MAJOR FINDINGS	COMMENTS AND RECOMMENDATIONS
Causes of Boiler and Pressure Vessel Accidents. December, 1984	Jagger, R.	National Board of Boiler and Pressure Vessel Inspectors' survey. Chemical	1984	-of the general problem areas which lead to accidents, poor maintenance and testing of controls ranked first, well ahead of most other categories. -within the category of poor maintenance and testing of controls, safety valve failures ranked highest and were observed to occur "quite frequently." -regular testing programs for safety devices are not being followed.	Design and construction cures are dependent upon the formulation and adoption of good construction and installation codes and standards. Equally important is regular third party inspection by the National Board.

TABLE 4: FAILURE MODES/MAINTENANCE TIPS

TITLE	AUTHORS	DATE OF PUBLICATION	FINDINGS AND COMMENTS
Halting Product Loss Through Safety Relief Valves	Bright, G.F.	June 1972	- when maximum allowable working pressure is close to normal operating pressure frequent popping of the relief valve may occur due to relatively small variations in operating pressure. - if valve seats are damaged from either corrosion or blowing foreign matter through the valve, leakage can be stopped by either selection of an appropriate resilient seated valve or by use of a rupture disc installed under the relief valve. - when a SRV opens and fails to close off: 1) dirt may have become lodged between the guiding surfaces of the valve, galling these surfaces, and holding the valve fully or partially open. A good solution is to investigate the use of bellows sealed SRVs in which foreign matter from both inlet and outlet of valve is isolated from guiding surfaces. 2) chattering will gall in the building surfaces and eventually lock up in a fixed position, refusing to either open or close. A chattering valve is not safe even if it does not gall, as its capacity is only a fraction, possibly 25-30% of the rated capacity of the valve.
The Certification and Type Selection of Safety Valves	Thornton, B.	1979	- a loss of some portion of the set pressure (after a period in the plant) is the most prevalent mode of failure. This is due to: 1) loss of loading force due to corrosion of spring; 2) loss of loading force due to settlement of the spring loading points because of corrosion or deformation; 3) increase in the sealing diameter between the valve seat and disc, due to corrosion or previous valve action. - in instances where the set pressure increases, this is usually due to adhesion of the seating or guide surfaces due to corrosion or deposits of solid matter on the moving parts
Design of Spring Loaded Safety Valves	Sharma, D.K.	April 1981	- an essential feature of the disc-seat interface is to ensure the point of thrust of the valve spindle is located somewhat below the seating faces. This feature of the disc-seat interface tends to reduce the valve chattering due to self-righting tendencies of the disc - knife edge seat valve provides precise popping, closing and leak-tightness characteristics when compared to the theoretical opening pressure due to the downward load acting on the disc. - 90 degree seats (flat seats) are preferred due to the easy grinding-lapping operation and unhindered movement across the flat surface without jamming on thermal expansion.
Safety-Relief-Valve Malfunctions: Symptoms, Causes and Cures o continued on next page	Scully, M.A.	August 1981	- SRV malfunctions are considered under three main headings: leakage, chatter, and premature popping - LEAKAGE: 1) operating pressures approach valve setting. Cure: use O-ring pressure seals. 2) corrosion or erosion 3) popping tolerance- when operating pressure is high, a pressure surge can unseat the disk and cause leakage. 4) solid particles between seat and disc - special knife-edge seat arrangement may cut through deposits or particles. For small entrained particles that enter close fitting-tolerance guides, jamming the SRV open use bellows to isolate guides from fluid. 5) vibration of vessel. When operating pressure is near valve set-pressure, vibration will make valve tend to leak. Cures: increase differential between operating and set pressure. 6) incorrect assembly: valve may seem to be jammed or stuck open when set for "long" blowdown - due to incorrect maintenance. 7) incorrect lapping of seats during renovations 8) testing errors or omissions: valves should be retested before being installed. Valves should never be installed in the as received condition. An estimated 2 to 3% of all valves shipped become misaligned.

TITLE	AUTHORS	DATE OF PUBLICATION	FINDINGS AND COMMENTS
Safety-Relief-Valve Malfunctions: Symptoms, Causes and Cures o *continued from last page*	Scully, W.A.	August 1981	- CHATTER: 1) oversized valves. At popping pressure there will be insufficient energy in the vessel to overcome the valve-spring force and lift the disk fully off the seat. - PREMATURE POPPING: 1) avoid making internal adjustments when there is pressure under the valve disk. 2) check for temperature correction when cold-setting. 3) check for incorrect reassembly of valve. 3) check for inadequate test facilities.
Safety Valves: a Friend in Need...	Garvin, W.L.	November 1981	- if a safety valve or SRV is subjected to pressure at or near its set pressure, it will tend to weep or simmer, and deposits can accumulate in the seat and disc area. Eventually this can cause the valve to freeze closed and thereafter the valve could fail to open at the set pressure. - test facilities may not have sufficient capacity to accurately verify the blowdown setting. The setting cannot be considered accurate unless made in the field on an actual installation - no matter how much quality and reliability is built into safety or safety relief valves, these devices can never be forgotten once installed and placed in service. SRVs require periodic inspection, and maintenance by properly qualified persons.
Ten Steps to Upgrade Safety-Valve Choice, Maintenance	Bordelon, T.R.	October 1983	- an organized program, divided into steps, is presented for engineering and maintenance supervisors to be used when planning inplant protection: - STEPS: 1) VALVE SELECTION: consider service, set pressure; SV testing should be at full overpressure conditions. 2) PREINSTALLATION: store in a dry, safe location and in vertical position protecting against bumping and mishandling. 3) INSTALLATION: make a final inspection. 4) INLET PIPING: should be as short as possible. 5) VALVE CONNECTIONS: use proper bolting materials and gaskets. Centre gasket to prevent flow restrictions. 6) REACTION FORCES: anchor properly and inlet necks and nozzle attachments must be able to absorb high loads because of dynamic loads occurring during rapid valve opening and closing. 7) ACCESSIBILITY: to ease maintenance. 8) ENVIRONMENTAL PROTECTION: eg. corrosion or springs must be considered. 9) MAINTENANCE: a preventative maintenance program is most important of the several key factors here. Procedures for periodic inspection, adjustment, or repair are the best way to have safe reliable valves. Performance testing is another vital aspect of valve maintenance. 10. PROPER VALVE CARE: periodic disassembly and visual inspection of all components.
Benefit from Systems Approach With Today's Safety Valves.	O'Keefe, Wm.	January 1984	- factors that might prevent achievement of full lift could be excessive friction on guide surfaces, excessive deposits or corrosion products along guide surfaces and large differences in the fluid conditions encountered. - periodic inspection should include the valve-body interior because the body itself is subject to erosion from steam and water droplets. - spindles, seats and guide surfaces can be corroded, bent, or scratched, and the setting may change because of rough handling or careless storage.
Care Needed in Repairing Safety Valves o *continued on next page*	Harrison, S.F., Jr.	March 1984	- possible effects of improper repair: most importantly: reduction of the valve relieving capacity. Other effects: shift in valve's set pressure, or blowdown outside the specified tolerances of ASME codes. Improper repair may cause movable components to hang up, flutter, or chatter. - Improper repairs are classified into two categories. These are: 1) that caused by workmanship, and 2) identifying deficiencies.

TABLE 4: FAILURE MODES/MAINTENANCE TIPS (CONTINUED)

TITLE	AUTHORS	DATE OF PUBLICATION	FINDINGS AND COMMENTS
Care Needed in Repairing Safety Valves o continued from last page	Harrison, S.F., Jr.	March 1984	- WORKMANSHIP: category includes overall mishandling of valve and components. Eg. overtighting valve in a vise during disassembly; cracking of body or deforming of a part might result in misalignment or binding once valve is reassembled; supporting valve by its lift can even cause damage to valve spindle by elongating or bending it; inadequate preparation for shipment: includes mounting horizontally, insufficient packing material, absence of covers over inlet. Fabricating replacement parts: quite frequently replacement parts are not manufactured to original manufacturers specs. Improper remachining of worn or damaged components. Installation of incorrect springs restrict valve lift and induce inability to meet set pressure and blowdown specifications and reduce valve relieving capacity. - IDENTIFYING DEFICIENCIES: category includes the failure to identify and correct deficiencies in one or more components of a safety valve. One of the most commonly overlooked items of this nature is cracked components. Also, failure to identify a spring that may be contained in a valve that does not meet manufacturer's specifications. Commonly overlooked items include untrue or bent spindles. Even a small amount of untrueness can prevent a valve from attaining its full lift and rated capacity.
Pressure-Relieving Systems	Moore, A.	October 1984	- valve stability depends in part on valve design but also on its installation and adjustment. To prevent chatter, or generally unstable or cyclical operation of the valve, the pressure drop should usually be less than 3% of set pressure, set pressure should be at least 10% above normal working pressure, and the minimum blowdown should be between 2 1/2 and 5% below the set pressure.
Special Report: Monitoring Valve, Actuator Performance	O'Keefe, Wm.	July 1988	- traditional safety valves have practically no margin of closing force available at the set point. There, the opening force on the disc is just balanced by the spring. Minor and unavoidable tremors in steam pressure will lift the disk very slightly to allow a small amount of steam to escape. As the minor pressure surge fades, the disc closes again. Rapid repetition of this causes simmering which causes damage to the sealing surfaces. - transportation of SVs has often caused problems, because of disturbance of set point and regulation of valve movement.

4.1.3 RUPTURE DISCS AS PRESSURE RELIEVING DEVICES

As with section 4.1.2, this section is considered in two parts. In Table 7, the experimental studies which have been conducted on rupture discs are described. Table 8 describes various papers which discuss the advantages and disadvantages of rupture discs, pressure relief valves, and the combination when used both in series and in parallel.

A limited number of experimental studies have been conducted on rupture discs during the last decade. Test areas have included: 1) optimum pre-doming of diaphragms; 2) effects of rate of pressurization on rupture disc performance; and 3) shock effects in relief manifolds consisting of two or more discs in series. The results of a quality control program in the Netherlands are also discussed.

As Baha et al (1979) explained, the lower the value of the hoop stresses operating at the dome of a diaphragm, the greater are the creep and fatigue strengths of the diaphragm. Baha et al (1979) determined this optimum predoming for metallic safety rupture diaphragms subjected to hydrostatic pressure. It is shown that hoop stresses are at a minimum when the ratio of dome height, h, to semi-chord length, a, has a value of $1/\sqrt{3}$. At optimum pre-doming, the value of the hoop stresses was found to be equal to:

$$0.966 \times \frac{\text{SEMI-CHORD LENGTH, a}}{\text{INITIAL SHEET THICKNESS, t}} \times \text{WORKING PRESSURE.}$$

Figure 8 shows the variation in hoop stresses with dome height.

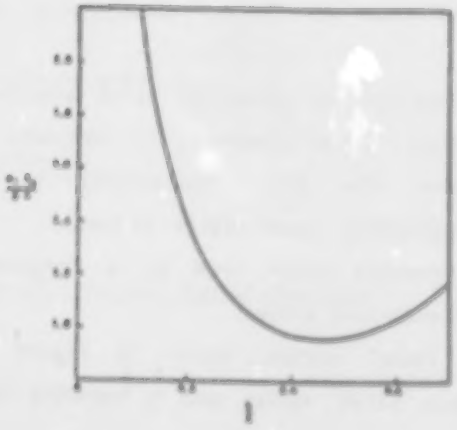


Figure 8

Variation of Operating Hoop Stresses with Dome Height

From Baha et al (1979)

In order to achieve a potential savings of material used to construct the vessel, Baha et al (1979) suggested that it was better to dome to 92% of the specified bursting pressure of the disc even when this was beyond the optimum doming position. The increase in the value of the hoop stresses operating at the dome of the diaphragm are at most about 2% and they have a negligible influence on the long term performance of the diaphragm.

Prickett (1984) investigated the effect that the rate of pressurization has upon the deformation and bursting characteristics of aluminum discs. He initially confirmed that two distinct deformation mechanisms arise at different rates of pressurization. For slowly applied uniform pressure, there is initially an elastic deformation. Increasing pressure then causes the disc to deform plastically. The plastic deformation begins at the centre of the disc, and further increases in pressure cause the elastic-plastic boundary to move outwards towards the rim of the disc. As the disc continues to deform plastically, the disc becomes thinner. At the bursting pressure the stresses at the centre of the disc cause the material to fail and the disc ruptures. When a disc is subjected to a very fast pressure rise the deformation process is entirely different. After the disc is stressed to its elastic limit an elastic-plastic region is created at the rim of the disc. The elastic-plastic discontinuity is designated as a hinge, and the position of this hinge at any time during its travel is designated as the hinge circle. The radius of this hinge circle decreases at a rate dependent upon the mechanical properties of the material and the pressure rise rate. The material within the hinge circle remains virtually unaffected until the hinge passes over it. The disc ruptures when the strain at the hinge circle is sufficiently large. Figure 9 contrasts the two types of deformation observed during quasi-static and dynamic pressurization of a disc.

The results of the deformation and bursting of 0.43 mm thick aluminum discs are presented in Figure 10. Prickett (1984) observed the increase in the bursting pressure required to burst the discs dynamically to be approximately five percent of the corresponding quasi-static pressure. He concluded that this increased bursting pressure could lead to a dangerous delay in the venting of the contents of a pressurized system.

Beveridge et al (1984) studied the relief system, shown in Figure 11, which consists of a manifold with multiple relief valves and a bursting disc

Figure 9: Typical Dome Profiles for: a) Quasi-Static; and b) Dynamic Loading.

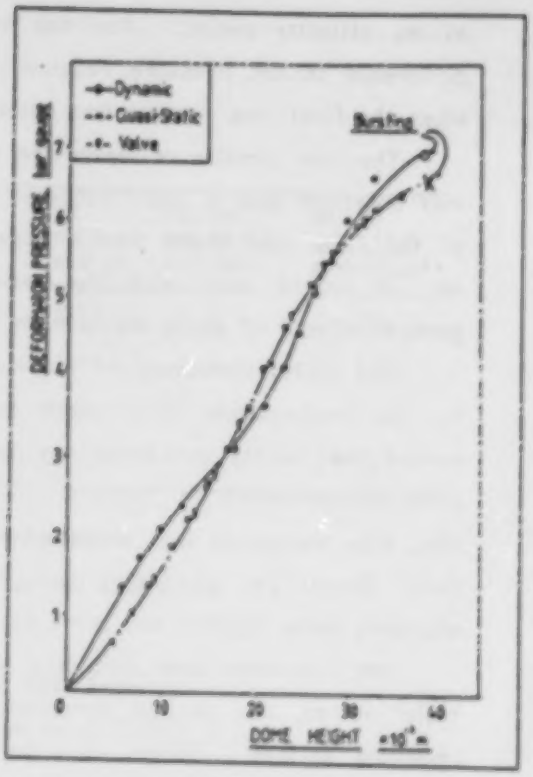


Figure 10: The Deformation and Bursting of 0.43 mm Thick Aluminum Disks

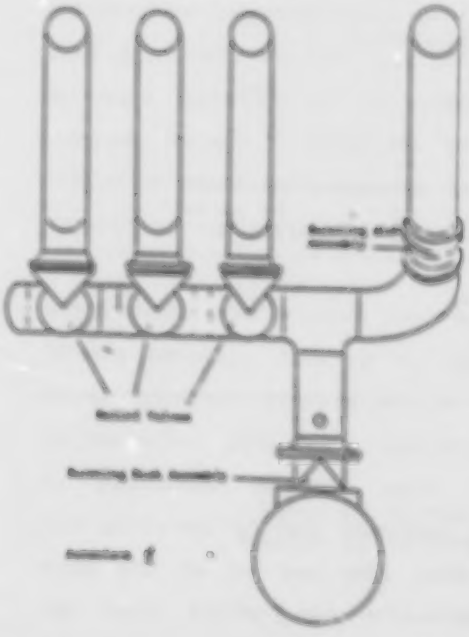


Figure 11: Typical Relief Manifold

Figures 9 and 10: Adopted from Prickett (1984)

Figure 11: Adopted from Beveridge et al (1984)

as an ultimate relief. For the two bursting discs in series, the minimum difference in set pressure required to allow the second disc to remain intact when the first was rupture was determined.

The test results of Beveridge et al (1984) are presented in Table 5. It was observed that if the first disc failed at or above 77% of the set pressure of the final disc there was a chance that the latter would rupture. Results will of course vary with the relief manifold arrangement employed but the general effects of shock waves have been identified.

Ens (1979) comments on the rupture disc quality control program employed by the Netherlands Government Boiler and Pressure Vessel Inspectorate to ensure that safety standards are met. The results for the years 1975 through 1978 are presented in Table 6. The high percentages of rejected batches at that time suggested that manufacturers' standards required improvement. Since then, Zanetti et al (1984) has stated that manufacturers are turning out sturdier, more reliable and more durable discs.

Many authors have discussed the pros and cons of rupture discs, pressure relief valves, and various combinations of the two when employed as pressure relieving devices. Others have concentrated solely on rupture discs and have discussed the various advantages and disadvantages of the different types of discs and in the process, they have identified the state-of-the-art devices. All of these papers, and their conclusions are presented in Table 8. This section continues by highlighting some of the most interesting information contained in these papers.

Kayser (1972) and Mathews (1983) are the only authors who identify the possibility of using pressure relieving devices in parallel. Kayser (1972) suggests that a safety relief valve be used as the primary relieving device with a rupture disc being employed as a supplement. The disc, being set at the higher relieving pressure, only opens when extreme conditions of overpressure are encountered. Mathews (1983) suggested that by installing two pressure relief devices in parallel, provided that they are set at the same pressure and that an interlocking isolation valve can ensure that one protective device is always on line, the difficulty associated with device replacement is drastically reduced. Other than these two articles, all of the literature considered rupture discs in series with safety relief valves.

The advantages and disadvantages of the different relieving-device-

TRIAL NUMBER	NO. OF TESTS	AVE. % BP 1 BP 2	AVE BP 1 (BAR A)	STD DEV	AVE BP 2 (BAR A)	NO. OF 2ND DISKS FAILING*	NO. OF 2ND DISKS SURVIVING**
1	12	72.0%	16.91	0.63	23.49	0	12
2	12	76.7%	17.95	0.97	23.41	1	11
3	12	79.1%	18.77	0.52	23.72	11	1
4	12	83.0%	19.70	0.71	23.72***	12	0

* Burst (failed) simultaneously with the 1st disk.
** Survived the burst of the 1st disk.
*** Periodic check tests were carried out on 2nd disk alone.

Table 5: Summary of Test Results.
Adopted from Beveridge et al (1984)

	Number of batches tested	Number of batches approved	Number of batches rejected	percentage rejected
1975	688	530	158	23 %
1976	669	489	180	27 %
1977	664	514	130	20 %
1978	610	493	117	19 %

Table 6: Number of Rupture Disc Batches Tested, Number of Batches Approved, Number of Batches and Percentages Rejected.
Adopted from Ens (1979)

arrangements are best summarized by Brodie (1979), Harris (1983), Mathews (1983) and Zanetti et al (1984). It is suggested that by installing a rupture disc upstream of a relief valve the best characteristics of both devices are utilized. These advantages include: 1) leak tightness; 2) reduced maintenance costs; 3) simplified valve testing techniques; 4) increased valve life; and 5) by using lower quality trim material on the relief valve, costs are reduced. The reader is referred to Table 8, where these advantages and disadvantages are discussed in greater detail.

TABLE 7: RUPTURE DISC TESTS

TITLE	AUTHORS	DATE OF PUBLICATION	RESEARCH INTERESTS	FINDINGS	RECOMMENDED AREAS OF STUDY
Rupture Discs as Pressure Relief Devices	Ens, H.	1979	Destructive testing of rupture disks as required in the Netherlands to ensure safety standards are met.	- number of batches tested, and percentage of rejected batches are presented for the years 1975 through 1978. These results are presented in the text of this report. - testing results indicate that manufacturers must be confronted, and their standards improved.	No comment.
Optimum Pre-Doming of Metallic Safety-Rupture Diaphragms.	Baha, D.Y. Beese, J.G.	1979	To study the influence of predoming on the value of the hoop stresses operating in standard metallic rupture diaphrages when subjected to hydrostatic pressure.	1) Hoop stresses at the dome of the diaphragm created by the working pressure of the plant are smallest when the ratio of dome height to semi-chord length equals $1/\sqrt{6}$. The lower the value of the hoop stresses operating at the dome of the diaphragm, the greater the creep strength and fatigue strength. 2) It is better to dome to 92% of the specified bursting pressure of the disc even when this is beyond the optimum doming position. The increase in the hoop stresses is not significant and there is a potential savings of material.	No comment.
Shock Effects on a Bursting Disk in a Relief Manifold. o continued on next page	Beveridge, H.J.R. Jones, C.G.	1984	To determine the minimum difference in set pressure of two discs in series that will allow the second disc to remain intact when the first is ruptured.	1) If the first disk fails at or above about 77% of the set pressure of the final disk there is a chance that the latter would rupture.	No comment.

TABLE 7: RUPTURE DISC TESTS *(CONTINUED)*

TITLE	AUTHORS	DATE OF PUBLICATION	RESEARCH INTERESTS	FINDINGS	RECOMMENDED AREAS OF STUDY
Shock Effects on a Bursting Disk in a Relief Manifold. o *continued from last page*	Beveridge, W.J.R. Jones, C.G.	1984	All but one test were conducted on small scale.	Confirmation of results using full scale tests proved expensive. Thus, only one full scale test was used to confirm results found using small scale.	
The Performance of Bursting Discs at Varying Rates of Pressurization.	Prickett, P.W.	1984	To investigate the effect that rate of pressurization has upon the deformation and bursting characteristics of aluminum discs.	With dynamic loading there is an increase in the bursting pressure required to burst the discs. This increase is of the order of 5 % of bursting pressure under static conditions.	No comment.

TABLE 8: RUPTURE DISCS - PROS AND CONS

TITLE	AUTHORS	DATE OF PUBLICATION	FINDINGS AND COMMENTS
Pressure Relief Systems	Isaacs, H.	February 1971	- location of rupture disks upstream of relief valves guards against valve leakage in systems. Such an application also provides added protection in a system handling materials that are likely to cause fouling or corrosion of the relief valve to the extent of making it inoperative. - rupture discs have the following advantages over pressure-relief valves: no leakage until failure, and they are suitable for liquid slurry service.
Rupture Disc Selection	Kayser, D.S.	June 1972	- rupture discs can be used as a supplemental relieving device where relief valves are used as the primary. The discs are set at the higher relieving pressure as permitted by codes. The relief valve will open on mild overpressures, relieve a small amount of material, and reclose the process. The rupture disc will not function unless a more extreme condition arises. - installing a rupture disc upstream of a relief valve utilizes the best characteristics of both devices and allows for use of a lower grade downstream material.
Pressure-Relieving Devices	Anderson, F.E.	May, 1976	- rupture discs may be installed between a vessel and a spring-loaded valve, as long as there is a pressure gage, free vent or suitable indicator for rupture or leakage. - a table of general features, advantages, disadvantages and typical uses for pressure relieving devices is presented.
Safety Discs, A Protective Device for Safety Valves o continued on next page	Brodie, G.W.	1979	- discs mounted in series upstream of valves prevent the process media from contacting the valve internals until an overpressure situation occurs. This leads to reduced maintenance. Also, safety discs are leak free. - the normal recommended (working pressure) / (bursting pressure) ratio is 0.9 for reverse buckling discs. - advantages of safety valves with rupture discs in series: 1) system is positively sealed, preventing leakage of the product. 2) prevents the process products contacting the valve internals during normal operation. 3) since the product does not normally come into contact with the valve internals the condition of the valve and nose is preserved as is the condition in which it was fitted to the system. This allows for longer periods between necessary valve overhauls. 4) it is possible to test the valve without the need to dismantle from the plant. This reduces maintenance costs. 5) by preventing the media contacting the valve it is suggested that in some cases less expensive valves could be used. - the reason why the practice of fitting discs upstream of valves is only used to a limited extent is explained by the inadequacy of some discs to perform the task satisfactorily. - conventional or simple domed disc assemblies are not suitable for heavy duty pulsating conditions and/or at elevated temperatures. Fragmentation of some part of the disc membrane is possible. - composite slotted discs perform superior to conventional discs. They are designed to cause no fragmentation. - in reverse buckling discs, after the disc becomes inverted by pressure, the venting can be caused by one of four methods, depending on the design of the disc: 1) by impaling the inverting disc on rigid knife blades; 2) shearing along pre-formed grooves; 3) shearing the edge of the disc done on a suitable edge cutter; or 4) parting the disc from the holder.

TABLE 8: RUPTURE DISCS - PROS AND CONS *(CONTINUED)*

TITLE	AUTHORS	DATE OF PUBLICATION	FINDINGS AND COMMENTS
Safety Discs, A Protective Device for Safety Valves o *continued from last page*	Brodie, G.W.	1979	- the principal features required of a rupture disc in a rupture disc/safety valve combination can be summarized as follows: - discs to have a long life; - tolerance of bursting pressure to be as small as possible; - no disc fragmentation which could affect valve performance; - bursting pressure of a disc to be the same as the valve lift pressure; - venting capacity of disc to exceed that of valve; - capable of operating in gas/liquid conditions.
Select the Right Rupture Disk	Harris, L.R.	May, 1983	- bursting pressure of a rupture disk is very temperature-sensitive. - advantages of rupture disc/safety valve combination in series are: - zero leakage to the atmosphere; - longer periods between major valve overhauls; - valves can be tested in place; - less expensive valve trim material may be used; - valve life is extended by isolating corrosive fluids from internal valve parts. - advantages/disadvantages of safety valves on their own: - metal seated valves leak to the atmosphere; - high cost; - high maintenance required. Since process goes through the valve, it must be checked periodically for correct operation. The system must be shut down; - adjustable; - blow down - recloses after pressure relief; - reusable - recloses after each operation. - advantages/disadvantages of rupture discs on their own: - leak tight; - low cost; - low maintenance required; - fixed setting; - non-reclosing; - disposable - must be replaced after each operation.
Bursting Discs for Over-Pressure Protection o *continued on next page*	Mathews, T.	August 1983	- advantages/disadvantages of the different types of discs: - flat discs: - virtually impossible to predict the bursting pressure accurately so that a large margin is required between the operating and bursting pressures. - one exception is the brittle bursting disc (often made of resin-impregnated graphite). Particularly suitable for low pressures and corrosive environments. - conventional domed discs - can be difficult to provide and handle sheet material thin enough to allow small discs to burst at the lowest pressures. - even slight corrosion of the thin material often required will weaken it considerably. - fluctuating operating pressure too close to the rated bursting pressure will cause premature failure by fatigue of the highly-stressed material, or by creep, so that operating pressure has to be limited to about 70% of the rated bursting pressure, or less at elevated temperatures. - if disc fails to burst completely, the venting passage is obstructed and pieces sometimes detach themselves and may cause blockage downstream.

TABLE 8: RUPTURE DISCS - PROS AND CONS (*CONTINUED*)

TITLE	AUTHORS	DATE OF PUBLICATION	FINDINGS AND COMMENTS
Bursting Discs for Over-Pressure Protection o *continued from last page*	Mathews, T.	August 1983	- pre-weakened discs – by using a thicker disc weakened by cutting radial slits, for example, from the edge to a number of small holes near the centre, several problems can be overcome. - distortion creep and fatigue become less of a problem because the rest of the disc is considerably stronger than the metal between the adjacent holes is. - discs can be operated up to 80% of the rated bursting pressure. - slits help the disc to burst cleanly and fully, with much less likelihood of separation of fragments. - reverse buckling discs – a disc of this type is considerably thicker for a given bursting pressure. - it is necessary to arrange for the disc to be opened, and a widely-used arrangement is a sharp knife which pierces the disc as it flips over. Knife divides disc into petals which open fully to give an unobstructed vent passage. - the petals do not fragment, so that there is no trouble with pieces of metal flying downstream. - creep and fatigue is not a problem because stresses in the disc are small - any damage to the disc's smooth uniform spherical shape causes it to buckle seriously below its rated pressure. - bursting discs in association with safety valves: - disc prevents leakage of process materials, which might block the valve or prevent its opening, whilst the safety valve serves to re-close the vent if and when the pressure drops. - it is possible to test the set pressure of the safety valve by injecting gas between the valve and the upstream disc, without disturbing the process. - a danger with the disc/valve combination is that a pinhole in the disc can pressurise the space between the disc and the valve: the pressure at which the disc will burst is then the sum of its rated pressure and the pressure in the space between it and the valve. - it is permissable to instal two bursting discs or safety valves in parallel, with interlocking isolation valves to arrange that one protective device is always on line, whilst the other can be isolated to permit it to be serviced or replaced.
Good Breaks for Rupture Discs	Zanetti, R. Short, M.	October 1984	- some types of reverse-buckling devices are capable of handling both liquid and gas. There are quick change discs, which reduce drastically the downtime needed for disc replacement; and corrosion resistant discs. - advantages of mounting discs at the access to pressure relief valves are: - virtually leak tight disc acts as a solid metal barrier between vessel and valve. This prevents potentially harmful products from escaping, and saves product. - no downtime is needed to test the valve. Without the disc, the process must be shut down and the valve dismantled before testing. - performance and safety are enhanced because no product buildup adheres to the valves mechanical components. The disc also protects against corrosion, extends valve life and periods between overhauls, and allows use of less expensive valve materials. - conventional reverse-buckling discs have not been successful in liquid service – pressurized liquids deform the disc too gradually, with insufficient "snap" to provoke a sudden bursting. Attempts have been made to change this downfall. A new disc design is noted which has a C-shaped score line cut into it, plus a hinge that forces the device to crack along the C before it can become inverted. It is finding applications in rail tank cars.

TABLE 8: RUPTURE DISCS - PROS AND CONS *(CONTINUED)*

TITLE	AUTHORS	DATE OF PUBLICATION	FINDINGS AND COMMENTS
Are Designers Making the Most of New Bursting Disc Devices	Beveridge, H.	August 1987	Reverse buckling discs: - the compressive strength of the disc material is used to determine its bursting pressure. - relatively unaffected by fatigue, do not fragment and can be operated at 90% or more of their bursting pressure. - until the relatively recent development of scored reverse buckling discs, this equipment was not sufficiently reliable for use at high duty cycles. - discs are available that are formed and precision scored against an anvil and then annealed to restore the mechanical properties of the material. These new discs give a full unobstructed opening without fragmentation and bursts within an accuracy of 3%.

SECTION 4.2

MODELING STUDIES

4.2.1 VALVE DYNAMIC MODELING

As Stoves et al (1984) explained, without computer simulation the development of satisfactory pressure control equipment requires considerable experimenting which is expensive and time consuming. In comparison, computer simulation is inexpensive in time, manpower and facilities once the computer software is available. A most important and often overlooked advantage of simulation is its ability to investigate potentially dangerous or impracticable situations. Before computer simulation can be used to aid design, the performance requirements and system parameters which affect performance must be identified and incorporated within a design specification. To this end, many authors have attempted to develop dynamic models of spring loaded safety valves. These models are summarized in Table 9 and are discussed in greater detail here.

Dokainish et al (1978) developed a computer model to analyze the free and forced vibratory response of a relief system. The valve system consisted of a valve having a mass, resting on a seat having nonlinear spring characteristics, and retained by a helical spring. The response curves of the first and third harmonic and one-third subharmonic, the amplitudes of which vary with the frequency of the external force, were investigated. It was found that the harmonic, higher-harmonic, and subharmonic response of the system became unstable within certain frequency ranges and that the valve model exhibited free vibration characteristics of the "hardening spring" type in that its frequency of free vibration increased as the vibration amplitude increased. Dokainish et al (1978) concluded by recommending that actual PRV tests be performed to verify his analytical studies.

Using fundamental principals of rigid-body motion and fluid dynamics, Ray (1978) developed a semiempirical steady-state model of a PRV. He observed that the fluid inertia in the valve orifice produced a damping effect on valve motion; that the opening time of the valve was approximately proportional to

the ratio of orifice length to radius; and that the mass-spring combination could generate oscillations and chattering as a result of high frequency pressure vibrations at the valve inlet. Ray (1978) did not, however, compare any of his analytical results with actual experimental results.

By developing a one-dimensional model, Singh (1982) studied the effects of different parameters, such as spring-mass characteristics, geometry of internal parts, adjustment ring settings, and bellows, which influence PRV dynamic behaviour. The model, although based on a similar approach to that taken by Ray (1978), represented the fluid dynamic force acting on the valve disc in the form of an explicit analytical expression in an attempt to minimize empiricism. Singh (1982) observed that valve stability and duration of full opening could be increased by lowering guide adjustment rings, by using a softer spring, or by reducing backpressure effects. Stiffer springs, higher backpressures, and the lowering of nozzle adjustment rings led to instabilities and fluttering of the valve. Singh (1982) concluded by suggesting that further work needed to be done to calibrate the model against test data on actual PRVs.

Catalani (1984) studied the dynamic stability of PRVs and investigated the effects that various assistance devices have upon that stability. Main factors found to affect PRV performance were: 1) ring setting; 2) upstream fluid conditions (whether vapour or liquid discharge); and 3) upstream piping length. It was found that when the upper ring was positioned somewhat higher than its optimal setting the less favorable disk thrust could not cope well with the valve pressure drop. The stability was negatively affected by increased piping lengths. This was explained by the longer time delay needed to recover from the initial pressure drop on valve opening and the subsequent greater pressure decrease at valve inlet. With upstream saturated fluid conditions, although PRV dynamics were found qualitatively comparable to steam discharge conditions, the stability margin was reduced owing to stronger pressure fluctuations on opening. Under subcooled conditions, the PRV showed a degraded performance which resulted in high pressure accumulation due to the unfavorable force balance on the disc.

Catalani (1984) found that PRV performance was greatly improved by employing assistance devices. Devices considered were pneumatic dual assistance and electromagnetic assistance. Due to the addition of two forces

on the valve disk - the first one being a closing force and the second being an opening force - pneumatic assistance was found to lead to improved tightness even at pressure levels near the spring load opening set point, and enhanced valve stability on opening. The precision in opening and closing pressures were greatly improved. The electromagnetic force, being set point operated, created at a pre-established lift position an additional restraint on the valve stem that opposed any eventual further downward movement. This eliminated valve stem fluctuations and chattering and provided stable opening characteristics even on liquid discharge.

It is an accepted fact that PRV chatter, which is characterized by violent oscillation of the disk, is a dynamical phenomena representable by differential rather than algebraic equations. MacLeod (1985) explained that by determining the behaviour of the differential equations in the region of certain critical points, the conditions necessary for avoiding chatter could be determined. MacLeod (1985) investigated the stability by employing Routh-Hurwitz's criterion. He concluded by suggesting that the dynamic stability of a PRV depends on the pressure vessel, the piping system as well as the actual PRV characteristics, and that it is necessary to have an accurate set of differential equations for the complete system if an accurate analysis of stability is going to be completed.

TITLE	AUTHORS	RESEARCH INTERESTS	FINDINGS	RECOMMENDED AREAS OF STUDY
On the Nonlinear Response of a Relief Valve 1970	Dubninich, M.A. Sivadasy, M.M.	- to study the free and forced vibratory response of a relief valve subjected to dynamic loading.	- harmonic solution is not a sufficient approximation. Generation of higher harmonic terms is required. - harmonic, higher-harmonic, and subharmonic response of the system may become unstable within certain ranges of frequency. - the valve model's frequency of vibration increases as the vibration amplitude increases.	- actual relief valve tests should be performed to verify the analytical studies.
Dynamic modeling and simulation of a relief valve 1970	Ray, A.	- a nonlinear dynamic model of a relief valve is formulated from fundamental principles of rigid-body motion and fluid dynamics.	- fluid inertia in the valve orifice produces a damping effect on valve motion. - opening time of valve is approximately proportional to ratio of orifice length to radius. - mass-spring combination may generate oscillations and chattering as a result of high frequency pressure vibrations at the valve inlet.	- no comment
An Analytical Study of the Dynamics and Stability of a Spring Loaded Safety Valve 1982	Singh, A.	- to develop a one-dimensional model of a safety valve and to study the effects of different parameters which influence its dynamic behaviour.	- in general, valve stability and duration of full opening can be increased by lowering guide adjustment ring, softer spring or smaller backpressure effects. - a stiffer spring, higher back-pressure, or lowering of nozzle adjustment ring may lead to instabilities or fluttering of the valve.	- further work needs to be done to calibrate the model against test data on actual valves.

Dynamic Stability Analysis of Spring Loaded Safety Valves - Elements for Improved Performance Through Assistance Devices
 	
1984
	Catalani, L.	- to develop a computer model of a safety valve and study valve stability with and without pneumatic and electromagnetic assistance devices.	- for no assistance devices:
			  - more upstream piping reduces stability. Cause is longer time delay req'd to recover from initial pressure drop on valve opening.
			  - saturated fluid reduces stability margin (compared to steam) due to strong pressure fluctuations on opening.
			  - subcooled liquid reduces upward force on disc.
			- with assistance devices
			  - pneumatic assistance (2 additional forces: to open & close valve disc). Advantages: improved seat tightness near set opening pt.; enhanced valve stability (from extra opening force); opening/closing precision improved.
			  - electromagnetic force (additional force on valve stem to oppose downward movement; set point operated). Advantages: valve stem fluctuations and chattering eliminated.
				- no comment

Safety Valve Dynamic Instability: An Analysis of Chatter 1985	MacLeod, G.	- to determine design criteria for valves which will allow chatter to be consistently avoided.	- by considering the roots of the characteristic eqns. valve stability can be determined. - dynamic stability of SV depends on the pressure vessel and piping system as well as valve characteristics. - it is usually possible to stabilize valve operation by damping valve disc motion or by increasing spring stiffness. Both of these result in slower valve opening.	- no comment

TABLE 2: Survey of Valve Dynamic Models

4.2.2 SOURCE MODELING

Singh (1982) suggested that since valve dynamic performance depends strongly upon upstream and downstream thermal-hydraulic conditions it is necessary for any given valve application to couple interactions between the valve and upstream and downstream system configurations. The systems which have been modeled to date are outlined in Table 13 and are discussed below.

As part of a study of PRV sizing techniques conducted by a special ad hoc committee for the AAR Tank Car Committee (1983), a computer model of a tank car subjected to a high heat flux was created. The code was coupled with a PRV model. The PRV model, however, lacked an indepth analysis of the dynamic phenomena occurring during the venting process. The model assumed ramp functions for both opening and closing characteristics. Because of the simplicity of the model, it is not given indepth consideration in the current section.

In order to study the effect of different upstream pressure conditions on valve performance, Singh (1982) assumed a time varying pressure boundary condition at the valve inlet. He prescribed the upstream reservoir pressure to rise at a constant rate, peak, and then decrease uniformly. Ray (1978) took a similar approach by assuming a step increase in the valve inlet pressure. Catalani (1984) coupled his valve model with upstream piping and concluded, as is discussed in section 4.2.1, that valve stability is dependent upon pipe length.

Other, more complex systems have been developed within the last few years. A model developed by Taras'ev et al (1985), assumed a system to operate in the following manner. On the opening of a cut-off valve, compressed air passed from a receiver through a regulating valve and into an enclosed tank on which a safety valve was mounted. The safety valve used in the model was developed from empirical relationships of pressure and flow-rate factors obtained during experimental studies similar to the tests discussed in section 4.1.1. The results obtained using this model suggested that further refinements were required. Taras'ev et al (1985) suggested that the methods employed to compute the nonsteady flow of gas in the intake and discharge lines were the cause of the discrepancies observed.

Taras'ev et al (1987) developed a package of programs for the calculation of nonsteady-state processes taking place as a result of high pressure in a

tank and of safety valve performance in a system consisting of a tank supplying a pipeline, a safety valve, and a delivery pipeline. He again developed his safety valve model by relying upon empirical relationships. The model could accept both compressible and incompressible fluids. One of the aims of Taras'ev et al (1987) was to estimate the effect of the supply pipeline's diameter on valve performance. It was discovered that with decreased pipeline diameters, the intensity of slide vibrations within the safety valve increased drastically. This was not caused by insufficient gas in the pipeline, but rather by nonsteady-state phenomena.

Groimes et al (1985) developed the SAFIRE (System Analysis for Integrated Relief Evaluation) code as part of the work being completed by the DIERS (Design Institute for Emergency Relief Systems) group. The program functionally describes the hydrodynamics of a storage vessel with an emergency relief system, and is capable of calculating the system characteristics (temperature and pressure response) prior to relief system actuation as well as the subsequent discharge through the emergency relief system. The code is extremely versatile and allows the user to make many decisions regarding vessel and vent configurations, flow regimes (some of which will be discussed in section 4.2.3), and the possibility of external heat addition. The possible configurations and flow regimes are shown in Tables 10 and 11, and Figures 12 and 13.

Table 12 shows the many currently available models for predicting the consequences of fire attack on pressure vessels. The table, adopted from Moodie (1988a), has been rised by the authors of the present report. Of these models, only one (SAFIRE), incorporates a spring-loaded pressure relief valve into the overall system. Most of the other models have assumed the pressure relieving device to be a rupture disc, fusible plug, or a relief valve which opens instantaneously. For this reason these other models have not been further discussed in this report.

Figure 12 Vessel configurations that can be input to SAFIRE.

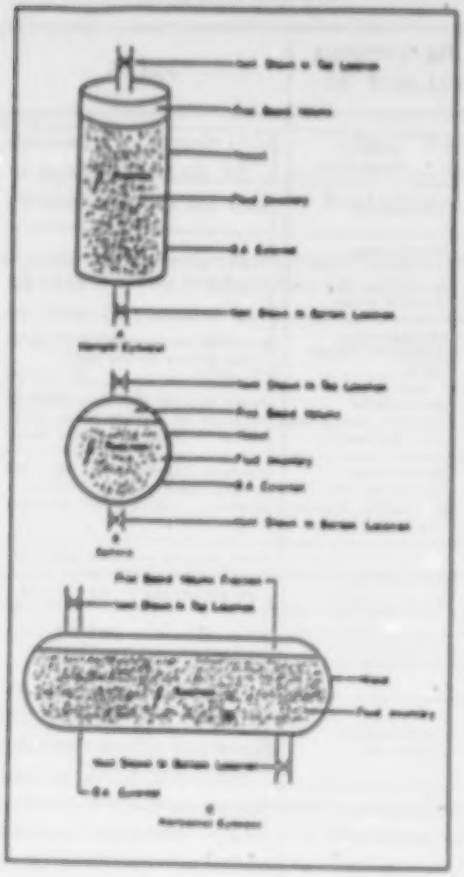


Figure 13 Vent flow configurations S1

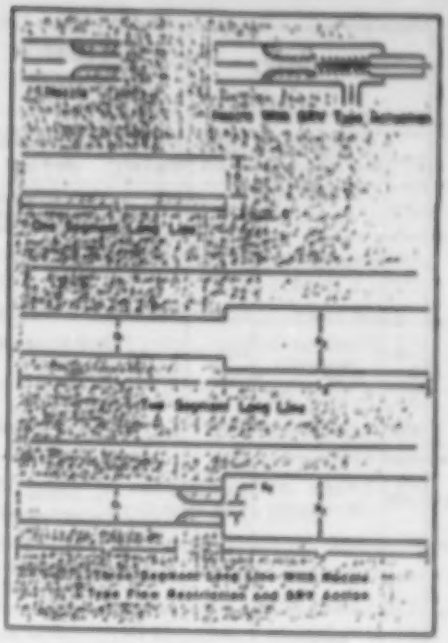


Table 10 Vessel flow regime and vent line flow model.

Vessel Flow Regime Selection
- Homogeneous mixture[1]
- Directed all-vapor flow[2]
- Directed all-liquid flow[3]
- Partial vapor-liquid phase separation[3,4]
  - Churn turbulent flow regime
  - Bubble flow regime

Vent Line Flow Model
- Directed all-vapor[5]
- Directed all-liquid (no flashing)[6]
- Nozzle vent
  - Equilibrium rate model
  - Homogeneous equilibrium flashing model
  - Frozen homogeneous model
- Long vent line models
  - Homogeneous equilibrium $10 < L/D < \infty$
  - Slip equilibrium $10 < L/D < \infty$ (1 segment only)

[1] Can be used for either top or bottom vent location.
[2] Top vent location only.
[3] Bottom vent location only.
[4] To be used with vessel type 1 only (Table 11).
[5] To be used in conjunction with corresponding vessel flow regime selection.
[6] To be used with bottom venting of a subcooled liquid.

Table 11 Vessel and vent configuration permissible in SAFIRE.

Vessel Type
- Right circular cylinder in vertical position
- Sphere
- Right circular cylinder in horizontal position

Vent Location
- Top or bottom

Vent Type
- Nozzle (friction effects are not important)
- Nozzle with SRV type actuation on pressure
- Long line (friction effects are important)
  - 1 segment
  - 2 segments
  - 3 segments (SRV type actuation)

Tables 10 and 11;
Figures 12 and 13.

All Adopted from
Croimes et al (1985)

Model Title	Date	Originating Body (Author)	Modes of fire attack modelled	Vessel Contents modelled	Vessel Type	Modes of Protection	Comments
-	1973	FRA (Giroux)	Engulfing fire	Pressurised liquified gases	Horizontal Cylinder	Uninsulated	Basic code in public domain
THICAR	1983	Queen's (Birk)	Engulfing fire	Propane	Horizontal Cylinder	Uninsulated, insulated, radiation shielding, Explosafe fusible plugs	Finite difference heat conduction three node thermodynamic model
ENGULF I	1984	HSE/SRD (Ramskill)	Engulfing fire	Liquified gases multi-comp.	Any	Uninsulated, insulated, water spray	A cuboid based model with three nodal points in the shell.
-	1984	UKB (Parmar)	Engulfing fire	Pressurised liquified gases	Horizontal cylinders	Uninsulated, internal protection inc. Explosafe.	First version based on CALSPAN modified to include valve interaction with heat transfer modes.
THICAR	1985	TDC (Birk/Davis)	Engulfing fire & 2D torch fire	Propane n-butane n-pentane propylene	Horizontal cylinder various orientations	Uninsulated, insulated, radiation shielding, Explosafe fusible plugs	Computer code predicts wall and lading temperatures, internal pressure, liquid level and time to failure
SAFIRE	1986	OIERD	Engulfing fire	Pressurised liquified gases	Any	Uninsulated	Wide ranging computer code for relief vent sizing which also includes external heat source
ENGULF II	1987	HSE/SRD (Ramskill)	Partial and full Engulfing fire & jet flames	Liquified gases multi-comp. gases	Horizontal cylinder vertical cylinder	Uninsulated, insulated	Later version of ENGULF I in cylindrical coords, enhanced fire modelling.
HEAT-UP	1987	SHELL RESEARCH (Deligeos)	Engulfing fire	Pressurised liquified gases	Horizontal cylinder	Uninsulated, insulated	2-D computer code for horizontal cylindrical propane tanks, pressure relief valve operation is modelled accurately
-	1987	UKB (Soudal)	Engulfing fire	Pressurised liquified gases	Horizontal cylinders	Uninsulated	Current UKB Code

TABLE 12: CURRENTLY AVAILABLE MODELS FOR PREDICTING THE CONSEQUENCES OF FIRE ATTACK

ADOPTED FROM MOODIE (1988)
REVISED BY AUTHORS OF PRESENT REPORT

TABLE 13: SOURCE MODELING

TITLE	AUTHORS AND DATE OF PUBLICATION	MODELING SOURCE	COMMENTS
Dynamic modeling and simulation of a relief valve	Ray, Asok November 1978	A step increase in valve inlet pressure is assumed.	No comment.
An Analytical Study of the Dynamics and Stability of a Spring Loaded Safety Valve	Singh, Avtar October 1982	A time varying pressure boundary condition is prescribed at the valve inlet. Sample calculations are based on steam flow.	Transient fluid and thermodynamic conditions are a strong function of the upstream and downstream system configurations.
Dynamic Stability Analysis of Spring Loaded Safety Valves - Elements for Improved Valve Performance Through Assistance Device	Catalani, L. August 1984	The valve is coupled with upstream piping	Valve stability is dependent on length of upstream piping.
Code Method for Evaluating Integrated Relief Phenomena	Grolmes, M.A. Leung, J.C. August 1985	A vessel coupled with a SV. Variable vents are considered, as are relief situations. External heat addition is possible. All vapor, flashing fluid or homogeneous equilibrium flow models can be employed.	No comment.
Mathematical Modeling of the Performance of a Safety Valve.	Taras'ev, Yu. I. Mamontov, V.N. Eremeev, Yu. V. Krivosheev, A.G. August 1985	Model consists of a receiver through which gas passes through a regulating valve & into a tank on which a safety valve is mounted.	Methods that make it possible to compute the nonsteady flow of gas in the intake and discharge line require further refinements.

TABLE 13: SOURCE MODELING *(CONTINUED)*

TITLE	AUTHORS AND DATE OF PUBLICATION	MODELING SOURCE	COMMENTS
Numerical Study of the Effect of Supply and Delivery Pipeline Resistance on the Operation of a Safety Valve.	Taras'ev, Yu. L. Eremeev, Yu. V. Krivosheev, A.G. July 1987	Model consists of a tank which supplies a pipe line. This line is connected to the SV. A discharge line is also present. Pipes have variable diameters Compressible and incompressible fluids are are considered.	When supply line's diameter increases the intensity of vibrations in the SV decrease This results from the non-steady state phenomena in the pipe rather than insufficient gas flow.

4.2.3 FLOW MODELING

During the last decade, researchers at the University of Maryland, Department of Mechanical Engineering, supported by the U.S. Department of Transportation, conducted a detailed study of two-phase discharge from finite vessels. Other programs, such as that supported by the DIERS group have also reviewed many of the existing flow models during investigations of the two-phase vapour-liquid onset and disengagement dynamics and relief system hydrodynamics. The results of these investigations, both theoretical and experimental, have been summarized in Table 14 and will be discussed in greater detail here.

The prediction of mass flow rates when two-phase flow occurs is a difficult task. Several different approaches to this problem can be found in the literature, such as the work of Messena (1960), Fauske (1962), Moody (1965), and Levy (1965). As Sallet (1979a) suggests, the validity of any particular existing two-phase flow model depends upon the specific flow pattern at or near the narrowest flow cross section between the pressure vessel and the atmosphere. If the critical cross section does not occur within a simple geometrical shape such as a pipe or a venturi, but occurs in a more complex structure like a pressure valve, the description of the simplest two-phase flow type, namely homogeneous flow then appears to be the most feasible one. The majority of the investigations discussed here have been based upon the homogeneous model.

Sallet et al (1978b) experimentally investigated the unsteady two-phase depressurization of vessels through orifices and short pipes and compared the results with various theoretical flow models. It was found that homogeneous equilibrium models (HEMs) underpredict the mass flow rate for orifices and short pipes. Although non-equilibrium models and slip flow models were found generally more successful, no single model was found adequate for all quality regimes and nozzle geometries. The differences in flow rate observed between several of the models, however, did become relatively small at high qualities. The mass flow rates which occurred during the venting of Freon-12 from the pressure vessel through pipes of various lengths are shown in Figures 14 and 15. In these figures, the experimental results are compared to theoretically calculated mass flow rates assuming frozen and homogeneous equilibrium flow. Sallet et al (1979b) concluded that for very short, orifice-like venting

pipes, in which the fluid cannot come to equilibrium before exiting the vessel, the venting rates can be calculated based on a frozen-flow model. By contrast, for long tubes in which thermal equilibrium is reached, a homogeneous equilibrium model gives good results.

Sallet et al (1981b) investigated the flow of gases and vapour from a rail tank car to the atmosphere during the final phase of the venting process, when only vapour is in the tank. The major points which were addressed in the study were the non-steady flow due to the finite pressure reservoir and the effects on the blowdown process if the vapour is not assumed to be a perfect or ideal gas. The theoretical equations which were employed when trying to match theory with the experimental results were 1) the perfect gas equation with different compressibility factors; 2) the ideal gas equation (with varying specific heat ratios); 3) Van der Wall's equation; 4) the Benedict-Webb-Rubin equation; and 5) Starling's equation. It was discovered that Starling's equation, which is reputed to be the most accurate equation of state for propane in the literature to date, agrees well with the non-steady pressure and mass predictions of the perfect gas equation when a compressibility factor of 0.7567 is employed. Sallet et al (1981b) concluded by suggesting that the perfect gas equation eases computational requirements because of the possibility of emplicit solutions.

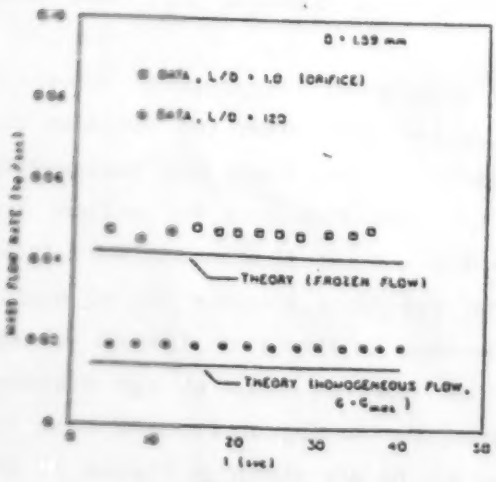


Figure 14

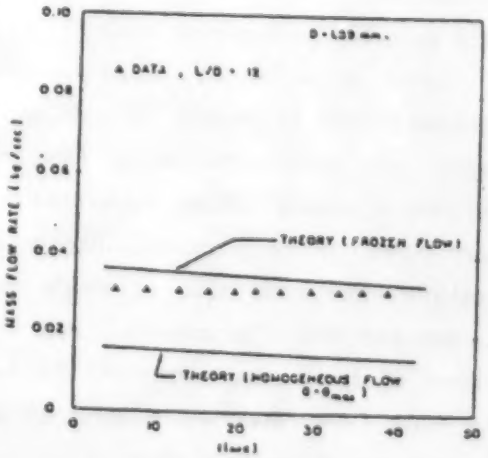


Figure 15

Figure 14: Mass Flow Rate Vs. Time (D = 1.59mm, L/D = 1,120)
Figure 15: Mass Flow Rate Vs. Time (D = 1.59 mm, L/D = 12)

Figures Adopted from Sallet et al (1978b)

Van den Akker et al (1984) developed a simple computer program for calculating homogeneous choked two-phase flow through pipes. The study compared the predictions of the program with results of vessel blowdown experiments with Freon-12 and water. The Freon-12 was used during small scale testing, while the water was utilized during the full-scale vessel blowdowns. It was observed that over the first 50 mm of the pipe the theoretical predictions of temperature were very close to the experimental results. Further downstream the experimental data and predicted temperature variations diverged to an increasing degree. Similar observations were made for the pressure within the pipe, where the theoretical results underestimated the actual pressure drop by up to 50%. The program did however correctly predict the blowdown rates for the small Freon-12 vessels. By means of a strongly increased value of the Fanning's friction coefficient a very close agreement was obtained between measured and calculated temperatures. However, after this modification was made it was observed that the discharge rates were being underpredicted by 30% or more.

Experimental investigations of the two-phase flow phenomena occurring during venting from rail tank cars were also carried out by Rod et al (1984). The set-up consisted of a small vessel with a square-edged orifice at the top of the chamber. The vessel was filled with Freon-12. System pressure and mass were continuously monitored throughout the proceedings. In all, 33 fully instrumented blowdowns were conducted. It was observed that the blowdown process is characterized by three distinct flow regimes. During the first one to two seconds of blowdowm the process is characterized by violent boiling and the establishment of a two-phase column. During the second regime, the exit flow is choked and may include entrained liquid. At this point the pressure and related properties fall at rates governed by the mass discharge rate. The third regime is characterized by subsonic flow. The system pressure has dropped, and the 2-phase column has subsided to a height that does not provide liquid entrainment to the exit flow. Here the mass discharge rate is governed by the heat transfer to the system from the environment.

Rod et al (1984) developed a computer program to model the blowdowm process. The model assumed that: 1) the fluid in the system and in the exit stream was always at thermal equilibrium at saturation; 2) the flow through the orifice was homogeneous flow with a slip factor of unity; and 3) the

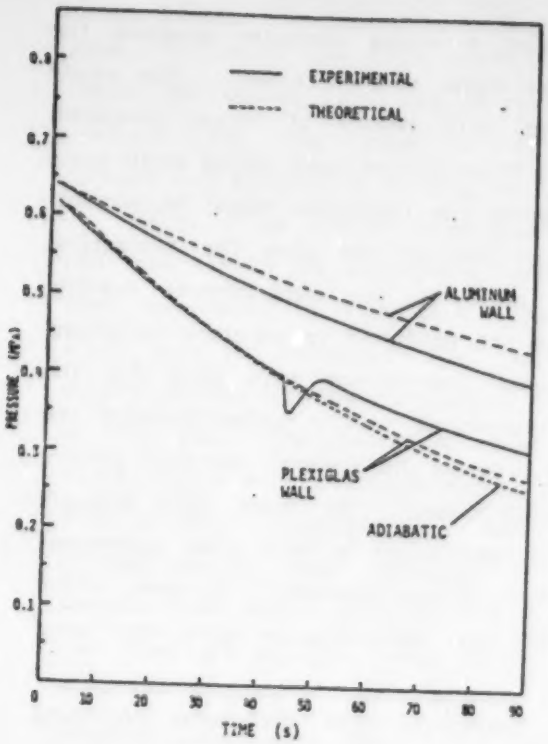


Figure 16

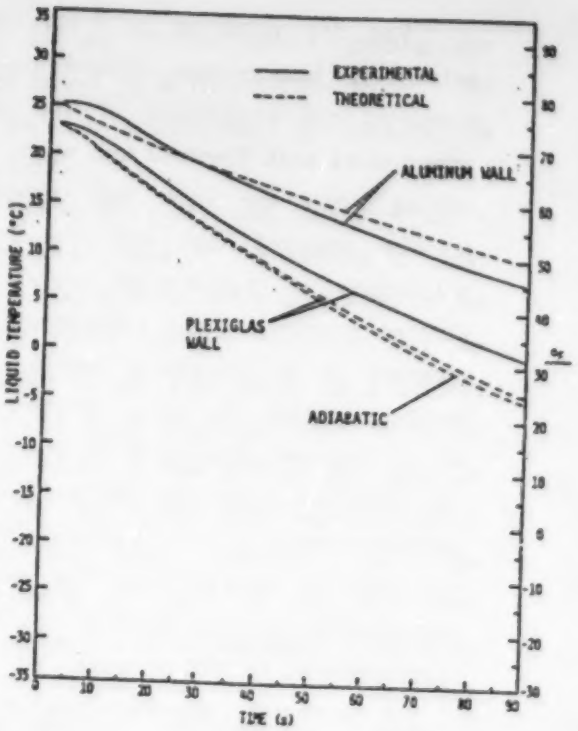


Figure 17

Figure 18

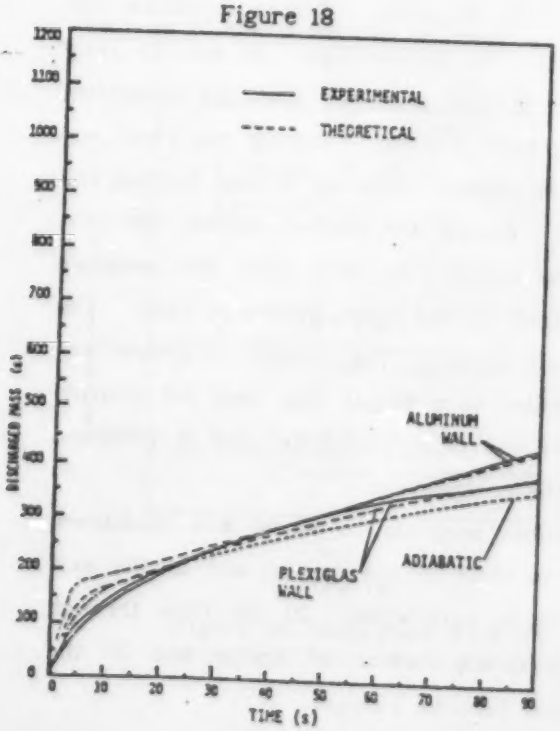


Figure 16: Theoretical and Experimental Pressure Decay (1.59 mm Orifice, 100% Liquid Fill Test.)

Figure 17: Theoretical and Experimental Temperature Decay (1.59 mm Orifice, 100% Liquid Fill Test.)

Figure 18: Theoretical and Experimental Mass Decay (1.59 mm Orifice, 100% Liquid Fill Test.)

All Figures Adopted from Rod et al (1984)

two-phase exit flow had a quality equal to that of the boiling column in the system. Comparisons of the theoretical and experimental pressure, temperature, and mass decays are shown in Figures 16, 17, and 18 for the 1.59 mm orifice, 100% liquid-filled-vessel tests. The basic heat transfer model employed appears to follow the experimental results fairly well for these tests. The model slightly underpredicts the thermodynamic state for the plexiglas wall test and overpredicts it for the aluminum wall test. The model accurately follows the experimentally measured mass discharge except for early in the two-phase discharge when the model's assumptions of homogeneity is most in error.

Huff (1985) discussed the flow models that will be presented in the DIERS project manual. After a brief discussion of the models considered during the project, he indicated that the end result of the exploration of alternate models was the selection of the homogeneous flow formulation as the most appropriate for general use in the DIERS program. He suggested that the model was quite simple to use and produced conservative results. Most importantly, the overall system model gave the best fit to blowdown data when the homogeneous model was used for the vent capacity predictions. He did however, suggest that when only the relief valve was being considered, better results could be obtained if some non-homogeneity was introduced into the model. This could be obtained by introducing a phase ship, or by assuming that the flashing process does not reach the equilibrium (saturated) conditions. One family of such models is based on the Henry-Fauske approach to non-equilibrium nozzle flow (Henry et al (1971)). The other approach to be presented in the DIERS Project Manual is a combination of the Schrock frozen flow model with the Burnell non-equilibrium flashing model (Sallet et al (1984)). These models are discussed further in the original sources.

Moodie et al (1987) investigated the venting characteristics of spring loaded pressure relief valves and the subsequent dispersion of the discharge during a series of fire engulfment tests. The results were compared with choked all-vapour flow predictions. For tanks which were either 60 or 80% filled, the mass flow rated during the initial stages of discharge were higher than those obtained from choked all vapour flow assessment. The observed rate of weight loss from the tank when 20% full was slightly less than the maximum vapour only prediction. The results of the investigation led Moodie et al

(1987) to conclude that for 5 tonne vessels, two-phase flow can be expected for initial fill levels greater than 50% of the vessel volume and when the heat flux to the liquid contents is in the region of 60 - 80 kW/m^2. For vessels smaller than 5 tonnes, two-phase flow occurs at fill levels of less than 50%.

TABLE 14: FLOW MODELING

TITLE AND DATE	AUTHORS	RESEARCH INTERESTS	DISCUSSION AND CONCLUSIONS	RECOMMENDATIONS
An Investigation Into Unsteady Two-Phase Depressurization of Vessels Through Orifices and Short Pipes 1978	Sallet, D.W. Hammann, R.J Gühler, M.	To study the depressurization of a vessel and to compare the results with various theoretical models	- homogeneous equilibrium models underpredict the mass flow rate for orifices and short pipes. Non-equilibrium models and slip flow models are generally more successful, although no single model is adequate for all quality regimes and nozzle geometries. - at high qualities the difference between the several available critical flow rate models is relatively small. - during experimentation the following were noticed: - constant mass flow rate over the first 60 seconds of the blowdown - critical flow rate increases with increasing tube length, due to increased quality. - for very short pipes, (L/D ≤ 1) calculations assuming frozen flow yield adequate results, while for longer pipes (L/D≥120), calculations should be based on homogeneous flow assumption	- observations made here apply only to one particular experimental set-up. Further experiments under various other conditions must be conducted before general conclusions may be drawn.
Two-Phase Flow Aspects in Sizing Pressure Relief Valves 1979	Sallet, D.W.	To review techniques employed to size relief valves and to investigate new techniques	- assumption of homogeneous flow in thermal equilibrium permits direct calculation of mass flow if critical flow rates are used - underlying assumption here is that flow is in thermal equilibrium. - the validity of this assumption will vary with valve position and valve design - full scale valve tests using a rail tank car safety valve & propane showed that the mass flow rates were well predicted by the critical homogeneous mass flux. - improved valve design and valve location may lead to non-equilibrium flow and, ∴ much higher mass flow rates.	No comment.
The Non-Steady Outflow of Propane Vapor from a Railroad Tank Car 1981	Sallet, D.W. Palmer, M.E.	To discuss flow of gases & vapor from a rail tank car to atmosphere during the final phase of the venting process when only vapor is in tank	- the perfect gas equation with a suitable compressibility factor is valid and fully sufficient for the flow of propane vapour. - Starling's equation agrees well with the non-steady pressure and mass predictions when the perfect gas equation with a compressibility factor of 0.7567 is employed.	No comment.

TABLE 14: FLOW MODELING (CONTINUED)

TITLE AND DATE	AUTHORS	RESEARCH INTERESTS	DISCUSSION AND CONCLUSIONS	RECOMMENDATIONS
Discharges of Saturated and Superheated Liquids from Pressure Vessels. Prediction of Homogeneous Choked Two-Phase Flow Through Pipes. 1984	van der Sikker, H. Bond, W.M.	To develop a computer program for calculating homogeneous choked two-phase flow through pipes. Predictions are compared with results of vessel blowdown experiments with freon 12 and water.	- predictions for temperature are close to experimental data over the initial 30 cm of pipe. - computer code in its present state underestimates temperature and pressure decay over most of the pipe length. - present model correctly predicts blowdown rates for small scale freon-12 vessels, but underestimates by some 50% the pressure and temperature drop over the blowdown pipe. - for full scale water discharge, temperature predictions are in much better agreement with experimental data (than freon tests); here the rate of discharge is underpredicted by some 25% - by strongly increasing the Fanning's friction coefficient a very close agreement can be constructed between the measured and calculated temperatures. However, the discharge rates are then underpredicted by 30% or more.	No comment.
An Experimental Investigation of the Discharge of a Flashing Liquid from a Finite Reservoir: A Fundamental Study of the Two-Phase Flow Phenomena Occurring During Venting from Rail Tank Cars 1980	Bud, S.R. Gühler, M. Dallet, D.V.	To study the process of blowdown from a rail tank car and to compare with theoretical models.	- the blowdown of pressurized freon from a test vessel can be divided into three regions: 1st region: - occurs for 1st one to ten seconds of blowdown. - characterized by violent boiling and establishment of 2-phase column. 2nd region: - either vapor flow or two-phase flow may exist in orifice. - distinguished by choked flow for its 2-phase equivalent which limits mass discharge rate. 3rd region: - after flow through orifice becomes subsonic - here mass discharge rate is governed by the heat transfer to the system from the environment. - a model is developed which assumes the following during blowdown: 1) system & exit stream fluid are at thermal equilibrium at saturation. 2) homogeneous flow through orifice with a slip ratio of unity. 3) 2 phase exit flow has quality equal to that of the boiling column in the system.	No comment.

TABLE 14. FLOW MODELING (CONTINUED)

TITLE AND DATE	AUTHORS	RESEARCH INTERESTS	DISCUSSION AND CONCLUSIONS	RECOMMENDATIONS
Multiphase Flashing Flow in Pressure Relief Systems 1989	Huff, J.E.	A discussion of the flow models to be presented in the DIERS project manual is discussed.	- the modeling problem with PRV's arises from the geometry complexity, as well as the varying degrees of non-equilibrium behaviour. - the assumption of no-slip flow with attainment of phase equilibrium in the flow to the nozzle throat (homogeneous equilibrium model, or "HEM") gives a conservative (low flow) result - better results are obtained by introducing some non-homogeneity into the model. - the choice among the flashing-flow models originally [illegible]	
Flow Through Pressure Relief Devices and the Dispersion of the Discharge 1987	Moodie, K. Jagger, S.F.	Venting characteristics of spring loaded PRVs and the subsequent dispersion of the discharge during a series of fire engulfment test are compared with choked all-vapor flow predictions	- in initial stages of discharge for 60 and 80% fills, the mass flow rate is higher than that obtained from choked all vapour flow assessment. - the observed rate of weight loss from the tank when 20% full was slightly less than the maximum vapour only prediction - test results indicate that two-phase flow is to be expected for initial fill levels greater than 50% of the vessel volume and when the heat flux to the liquid contents is in the region of 60 - 80 kW/m^2. - Two-phase flow occurs at less that 50% full for tests on vessels smaller than 5 tonne.	No comment.

4.2.4 EFFECTS OF TWO-PHASE FLOW

When the rise in pressure in a pressure vessel filled with liquefied gas is due to heat input from exposure to fire, the vaporization process within the tank may become so violent that the medium flowing through the valve is no longer a vapour but rather a mixture of vapour and entrained liquid. For this reason it is important that the effects of two-phase flow on relief valve performance be understood. Many recent studies have shown the effects of two-phase flow phenomena on valve performance. These experiments are summarized in Table 15 and are discussed in greater detail here.

Sallet et al (1985) obtained steam-water blowdown data that showed regions of safety valve instability. Large amplitude vibrations of the valve disc were observed when saturated or nearly saturated fluid flowed through the valve. Borzov et al (1986a) commented on similar observations when determining the two-phase hydrodynamic characteristics of a valve. The phenomena was only present with large lifting heights of the slide valve's seat. Borzov et al (1986a) noted a reduction of the stress exerted by the two-phase stream on the slide valve that amounted to approximately 30% of the stress exerted under conditions of liquid flow. The poor performance of the valve during two-phase blowdown led Sallet et al (1985) to conclude that cures for valve disc vibrations must be addressed.

Safety valves sized assuming vapour flow have proved inadequate under conditions of two-phase flow and much of the work of the DIERS group has been towards determining new sizing techniques. As Fisher (1985) suggested, with the same limitations on overpressure, two-phase venting normally requires between two and ten times the venting area of systems sized assuming single phase gas flow. He explained that a relief system extracts more energy from a vessel during all-vapour relief than it does during two-phase venting, and thus is more capable of moderating temperature (and hence pressure) rise during emergency situations.

Sallet (1982) and Sallet et al (1985) studied the effects of entrained liquid droplets on the flow of vapour through relief valves. Sallet (1982) found that while the combined mass rate of flow actually increased during liquid droplet entrainment, the mass rate of vapour flow decreased. He also observed that increases in droplet size tended to increase the combined mass rate of flow and that for a given liquid flow rate, the droplet size very

strongly influenced the pure vapour flow rate. In 1985, Sallet and Somers (Sallet et al (1985)) found the volume flow rate of saturated water to be much lower than that found for subcooled water. They also witnessed, as discussed previously, poor valve performance during the two-phase flow conditions.

For a more detailed description of the testing apparatus and conclusions of the papers presented here, the reader is referred to Section 4.1.1 where all of the valve tests have been discussed in greater detail.

TABLE 15: TWO-PHASE FLOW EFFECTS

TITLE	AUTHORS	DATE OF PUBLICATION	TYPE OF STUDY	DISCUSSION AND CONCLUSIONS	RECOMMENDATIONS
The Flow of Liquids and Gases Through Rail Tank Car Safety Valves	Sallet, D.W. Weske, J.R. Gahler, H.	July 1982	Experimental	- valve coefficients remain nearly the same when pure liquid flow changes to very low quality two-phase flow. - entrainment of liquid droplets decreased the mass rate of vapor flow, yet increased the combined mass rate of flow. - an increase in droplet size increased the combined mass rate of flow - for a given liquid flow rate the droplet size very strongly influenced the pure vapour flow rate with finer spray decreasing the vapour flow rates.	No comment.
DIERS Research Progress on Emergency Relief Systems	Fisher, H.G.	August 1985	Experimental/ Theoretical	- for the same limitations on overpressure, two-phase venting normally requires 2 to 10 times the vent areas as does single phase gas. - larger area required because an all-vapor relief system extracts more energy from a system than does two-phase venting. - two-phase venting extracts more mass from a system than does all vapor relief	No comment.
Flow Capacity and Response of Safety Relief Valves to Saturated Water Flow	Sallet, D.W. Somers, G.W.	October 1985	Experimental	- the volume flow rate of saturated water is much lower than that found for subcooled water. - for highly compressed liquid and saturated liquid restricting valve outlet does not decrease volume flow rate unless exit to nozzle area is 5 or less - valve discs experience large amplitude vibrations when saturated fluid was flowing - severe vibrations of valves with bellows occurred when exit to nozzle area was a 1.	No comment.

TABLE 15: TWO-PHASE FLOW EFFECTS (CONTINUED)

TITLE	AUTHORS	DATE OF PUBLICATION	TYPE OF STUDY	DISCUSSION AND CONCLUSIONS	RECOMMENDATIONS
Hydrodynamic Characteristics of Safety Valves for Two-Phase Heat Carriers	Borzov, B.A. Grishin, Yu. M. Kozlow, V.M.	April 1986	Experimental	- for valves with conical deflectors and with low lifting heights of the slide valve, the presence of moisture in the stream had practically no influence on the stress exerted by the stream on the slide valve. - for high lifting heights stress exerted by the two-phase stream on the slide valve is reduced by 30% in comparison with the single-phase stream. - passage of a two-phase stream through the seat slide valve section is characterized by sharp separation of the phases and the occurrence of a stratified flow regime.	No comment.

SECTION 4.3

CASE STUDIES

4.3.1 TANK FIRE ENGULFMENT TESTS

Since the mid-1970's, several investigations have been conducted to study the effects of fire engulfment on pressure vessels. During these investigations, instrumentation has allowed the authors to attain information regarding the transient pressures within the vessels. In some cases, the analysis of this data has suggested poor relief valve performance. Many have commented on the variable opening pressures of the relief valves and the fact that they often fail in the open position. Some of the more recent fire engulfment tests are summarized in Table 16 and are discussed in greater detail below.

Appleyard (1980) conducted a series of fire impingement tests while investigating the ability of EXPLOSAFE material, an expanded aluminum foil honeycomb-like structure, to reduce the hazard of Boiling Liquid Expanding Vapour Explosion (BLEVE) when LPG containers are exposed to fire. Figures 19 through 21 show the resulting pressure-history graphs obtained during three of the tests. Although each of the vessels employed during these three tests contained the EXPLOSAFE material, the pressure-time histories are very different. In both the DIABLO and DRES SPECIAL tests, Figures 19 and 20, the relief valves failed in the open position, causing all of the vessel-contents to be released to the environment. The deviation in the pressure-time graph for the DRES SPECIAL test between six and twelve minutes from ignition suggests that the relief valve was not operating in its usual cyclic manner. The change in the slope of the pressure-time graph for the NOVA test observed at one minute into the test indicates that the pressure relief valve cracked open for some unexplained reason. The relief valve appears to have remained in this cracked-open position until the set pressure was attained, at which time it began to cycle in the expected fashion. Appleyard (1980) suggested that the transient differences in the cycling behaviour of the relief valves

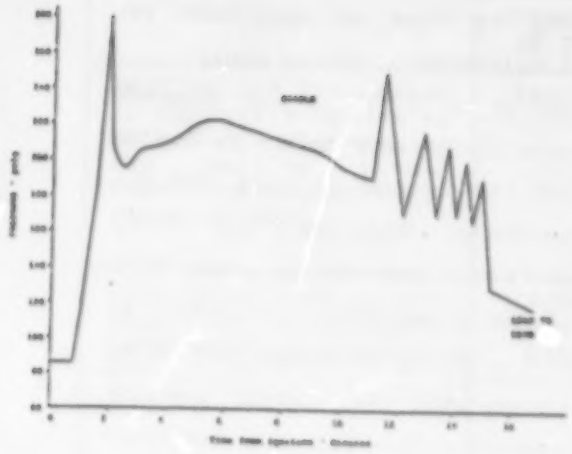


Figure 19

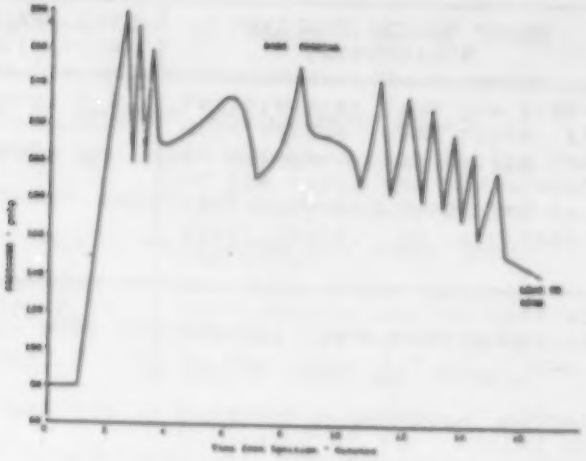


Figure 20

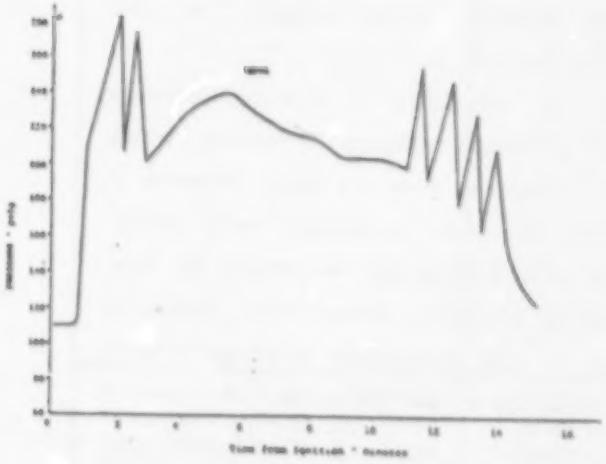


Figure 21

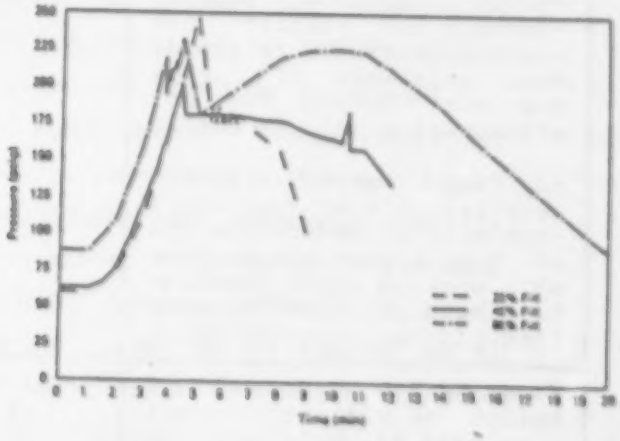


Figure 22

Figures 19-22: Pressure - Time Histories During Fire Impingement Testing

Figures 19-21: Adopted from Appleyard (1980)
Figure 22: Adopted from Moodie et al (1985)

were probably caused by the manufacturing variation in open/close set pressures and flow capacity coupled with small differences in fire intensity.

During investigations conducted by Moodie et al (1985), it was observed that in one of the five tests, the relief valve cycled once before failing in the closed position. The failure caused both the internal pressure and the vapour space wall temperature to rise uncontrollably. When the vessel finally ruptured, the internal pressure had reached 35 bar and the maximum wall temperature was approximately 600 °C. Moodie et al (1985) was unable to establish the cause of the relief valve failure. During the other four tests the relief valves remained in the open position until the internal pressure was eventually down to atmospheric or thereabouts. This fail safe behaviour was, upon subsequent examination of the valves, attributed to fire damage to the valve seats and to a weakening of the valve springs. Figure 22 shows the pressure-time history observed during the 1 tonne vessel tests. Although the relief valves had identical set pressures, initial openings ranged from 220 psig for the 80% fill test to 247 psig for the 20% fill test.

Roberts et al (1983) also observed the variability in opening pressure for relief valves. During testing, he witnessed opening pressures which ranged from 0.7 - 1.8 MPa when actual set pressures were 1.7 MPa. Roberts et al (1983) suggested that possible causes of this variability were either weakening of the spring or damage to the valve seating by the effects of fire. Moodie et al (1988b) observed initial opening pressures which were within 5% of set pressure for all tests. He observed that subsequent openings/closings were at progressively lower pressures, but gave no indication of possible causes.

TABLE 16: TANK FIRE ENGULFMENT TESTS

TITLE	AUTHORS AND DATE OF PUBLICATION	PRESSURE RELIEF VALVE PERFORMANCE
Testing and Evaluation of Explosafe System as Method of Controlling the Boiling Liquid Expanding Vapour Explosion (BLEVE).	Appleyard, R.D. 1980	Performance: from the pressure versus time plots it is observed that during one of the tests the SRV opened extremely slowly during the first cycle. No explanat- was given.
Fire engulfment trials with LPG tanks with a range of fire protection	Roberts, A.F. Cutler, D.P. Billinge, K. 1983	Performance: set to open at 1.7 MPa but actually opened in the range 0.7 - 1.8 MPa. Possible causes: either weakening of the spring or damage to the valve seating by the effects of fire.
The Assessment and Control of Major Hazards.	Moodie, K. Billinge, K. Cutler, D.P. 1985	Performance: in four of five tests they cycled at least twice and then remained open until internal pressure was approximately at atmospheric pressure. In remaining test, the PRV cycled once and failed to open subsequently Possible causes: for the first four test's failures are attributed to fire damaged valve seats and to weakened valve springs. No reasons for failure of fifth PRV were established.
The Use of Water Sprays to Protect Fire Engulfed LPG Storage Tanks.	Billinge, K. Moodie, K. Beckett, H. 1986	Performance: either performed correctly or failed by remaining in the open position without reclosing when the tank was depressurized. Possible causes: burnt valve seats prevented them from reseating.
Fire Engulfment tests on a 5 Tonne LPG Tank	Moodie, K. Cowley, L.T. Denny, R.B. Small, L.M. Williams, I. 1988	Performance: the first opening was always within 5 % of the set pressure. Subsequent openings/closings were at progressively lower pressures. Possible causes: N/A

SECTION 5.0

CONCLUSIONS AND RECOMMENDATIONS

5.0 CONCLUSIONS

This report has investigated the studies which have been conducted in the area of pressure relief technologies. The original focus of the report was to conduct a search of public domain literature to find the state-of-the-art for pressure relief valves (PRVs) which are presently being installed on railway tank cars, highway tankers, and stationary LPG storage vessels. When few papers were found which had concentrated on the topic, the authors continued the search by considering both the nuclear power and chemical processing industries, where similar technologies were found. Studies of particular significance in the research of pressure relief technologies have been the experimental work at the University of Maryland, under the direction of D.W. Sallet, and the research conducted by the Design Institute for Emergency Relief Systems (DIERS).

After screening all of the literature cited, that information deemed particularly relevant was placed into one of three categories: 1) experimental studies; 2) modeling studies; and 3) case studies. The conclusion section will continue by giving a synopsis of each of these three sections.

5.1 PRESSURE RELIEF TECHNOLOGIES

During the course of the literature search, it was discovered that the design of pressure relief valves currently installed on tank cars and highway tankers dates back more than 30 years. Other than such changes as different valve materials, the technologies have remained unchanged.

Rupture discs, on the other hand, have changed considerably. The once unpredictable devices have advanced to a state where they appear quite reliable. New devices, such as a disc capable of handling both liquid and gas and units that are resistant to large cyclic pressure variations, have taken the uncertainty out of their design. The devices are now gaining

acceptance in many industries.

5.2 EXPERIMENTAL STUDIES

The majority of the experimental research of the pressure relief valves (PRVs) installed on rail tank cars has been conducted at the University of Maryland under the direction of D.W. Sallet. Such topics as the flow patterns within the PRVs under both single and two-phase flow conditions, and tank depressurization phenomena have been addressed. Very few tests, however, have considered relief valves with discs which were free to move. In most case, the disc was set at a pre-established height. This limitation has prevented the obtaining of valve dynamic characteristics.

Investigations have shown that regular maintenance testing programs have not been followed in the past. Where limited testing has been conducted, in both the nuclear power and chemical processing industries, results have indicated that either many minor improvements in design, quality control, and application are needed, or else that fundamental rethinking is called for in PRV technologies.

Field experiences have shown that PRVs often fail to meet specifications. Frequently, the flow conditions within the valve are observed to result in chatter. This action can damage the valve, resulting in such effects as jamming, leakage, and the failure to open at set pressures. In some cases, even brief operation at the set point has been observed to cause damage to the PRV.

5.3 MODELING STUDIES

Few modeling studies of pressure relief valves (PRVs) have compared the analytical results obtained with actual test data. Those that have, continually observed a poor correlation between theoretical and experimental data. The authors have generally concluded their investigations by suggesting that further

modifications of the models were required.

A model developed by Grolmes et al (1985) is the only computer code which couples a spring-loaded pressure relief valve with a source appropriate for incidents involving tank cars and LPG storage vessels. The source, a pressurized cylinder subjected to a high heat flux, could be used to represent fire engulfment of a tank car. Most valve models have assumed a simplified time varying pressure distribution condition at the valve inlet.

The validation of the PRV dynamic models will not be possible until the operating characteristics of new and old PRVs are better understood.

5.4 CASE STUDIES

Very little information was found which related directly to accidents caused by relief valve failures. In the majority of cases, the incident which originally led to the rise in pressure was listed as the cause of the accident, and no mention was made of the relief valve performance.

The data which does exist on PRV performance during releases, obtained during controlled fire tests of tank cars, suggests that they often perform poorly. In the most serious of cases it has been observed that their malfunctioning has caused the violent rupture of the tank car.

Accidents should be investigated thoroughly. After an incident occurs, the PRV should be removed and tested extensively to determine whether it performed correctly, or whether it jammed or failed in some other manner. The results of such tests should be organized into a public domain data base.

5.5 RESEARCH NEEDS

During the course of this literature search the following conclusions have been drawn:

(1) Experimental research of valve operating characteristics are required.

(2) Maintenance and testing programs in both the nuclear and chemical industries appear inadequate. Current practices relating to the transportation of dangerous commodities should also be reviewed.

(3) Validation of existing dynamic computer models for PRVs is required.

(4) When an accident involving a pressurized vessel occurs, the PRV should be removed and tested extensively to determine whether or not it performed correctly.

(5) A data base should be established for accidents involving PRVs.

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